GEORGIA INSTITUTE OF TECHNOLOGY
OFFICE OF CONTRACT ADMINISTRATION
SPONSORED PROJECT INITIATION

Date: 9/25/79

Project Title: Investigation of Passive Cooling Techniques for Hot-Humid Climates

Project No: D-48-636

Project Director: J. M. Akridge

Sponsor: U. S. Department of Energy; Chicago Operations Office

Agreement Period: From 7/15/79 Until 9/30/82

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Sponsor Contact Person (s):

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Defense Priority Rating: None

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SPONSORED PROJECT TERMINATION SHEET

Date 1/14/83

Project Title: Investigation of Passive Cooling Techniques for Hot-Humid Climates

Project No: D-48-636

Project Director: J. M. Akridge

Sponsor: DOE

Effective Termination Date: 9/30/82

Clearance of Accounting Charges: 9/30/82

Grant/Contract Closeout Actions Remaining:

- [x] Final Invoice and Closing Documents
- [ ] Final Fiscal Report
- [x] Final Report of Inventions
- [ ] Govt. Property Inventory & Related Certificate
- [ ] Classified Material Certificate
- [ ] Other ______

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Progress Report No. 1

Progress Report No. 2

D-48-636

INVESTIGATION OF PASSIVE COOLING FOR HOT-HUMID CLIMATES

Progress Report
No. 3

Submitted to

Research and Development Branch
Heating and Cooling, Conservation and Solar Applications
Department of Energy

By the

College of Architecture
Georgia Institute of Technology
Atlanta, Georgia 30332

April 29, 1980

Project Director

James M. Akridge
Associate Professor of Architecture
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INVESTIGATION OF PASSIVE COOLING FOR HOT-HUMID CLIMATES

Work the past two months has been devoted toward getting the brookhaven National Laboratories computer program for modeling ground coil performance operational. This program proved to be much more difficult to get operational than had been anticipated. The problem was finally, with the aid of John Andrews of BNL, traced to a change in TRNSYS from the 8.2 version used by BNL to the 9.2 and 10.1 versions being used at Georgia Tech. John recommended several changes to the TRNSYS main program which were successful. The BNL ground coil simulation program is now operational on the Georgia Tech computer and is presently being used to optimize the field layout.

Arrangements have been made to sink the two 40 ft. vertical wells for soil temperature measurements. These wells should be sunk on May 5th. Figure 1 shows the well location relative to the field location. One well is located remote from the field to measure undisturbed soil temperatures and one well is located in the center of the field to measure the vertical effect of the field. Core samples will be taken during the drilling operation. These samples will be weighed, dryed, and reweighed to determine initial soil moisture content. Figure 2 is a crossection through one of the vertical wells. This figure shows the thermocouple location. The thermocouples are positioned in a sand filled vertical PVC tube with each thermocouple bead projecting through a small hole in the tube and being bonded to the exterior of the tube. Since the PVC tube has a thermal conductivity only 1/10 that of soil, the tube should not affect thermocouple measurements. Once the

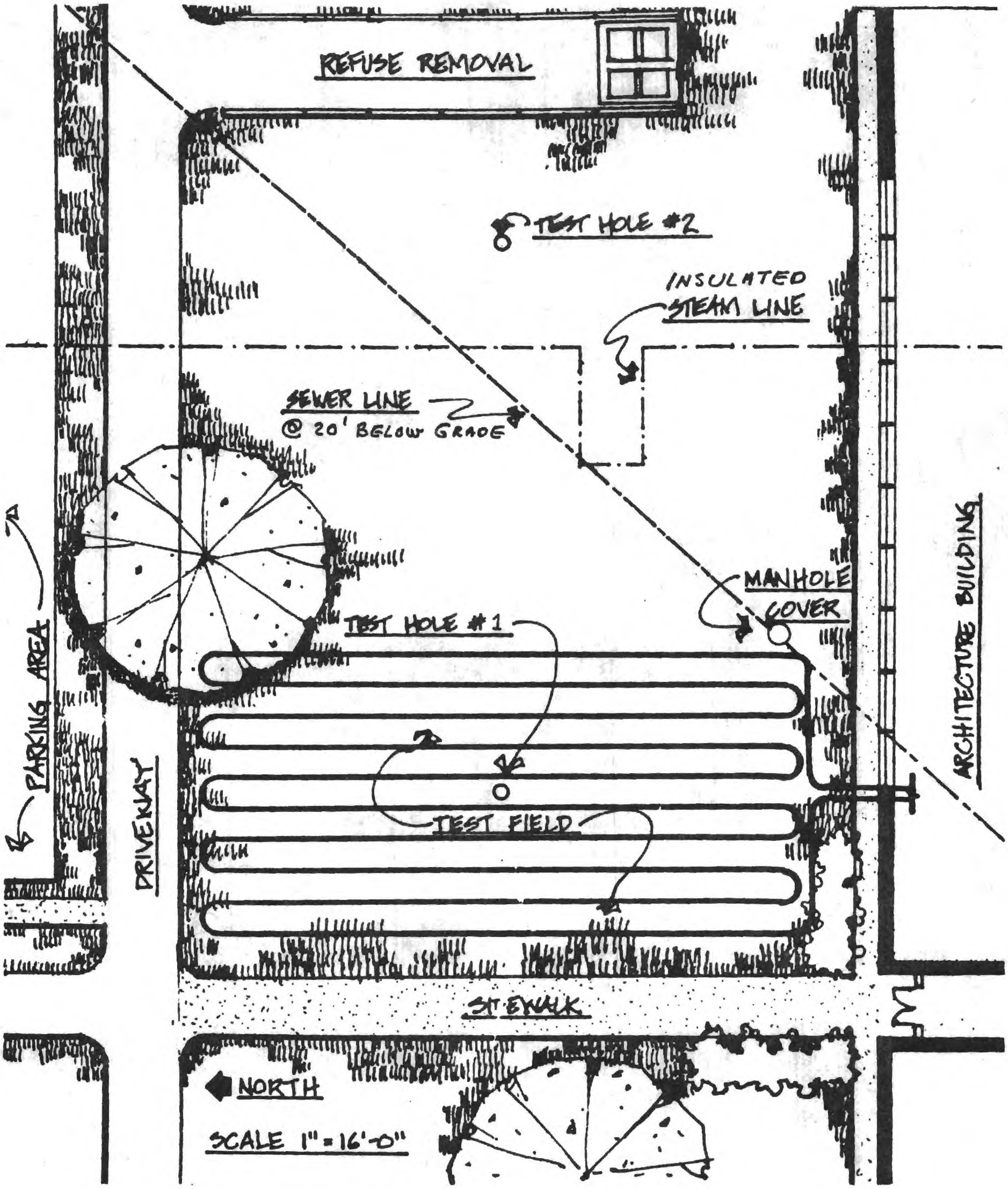

FIGURE NO. 1 Plan of Test Site

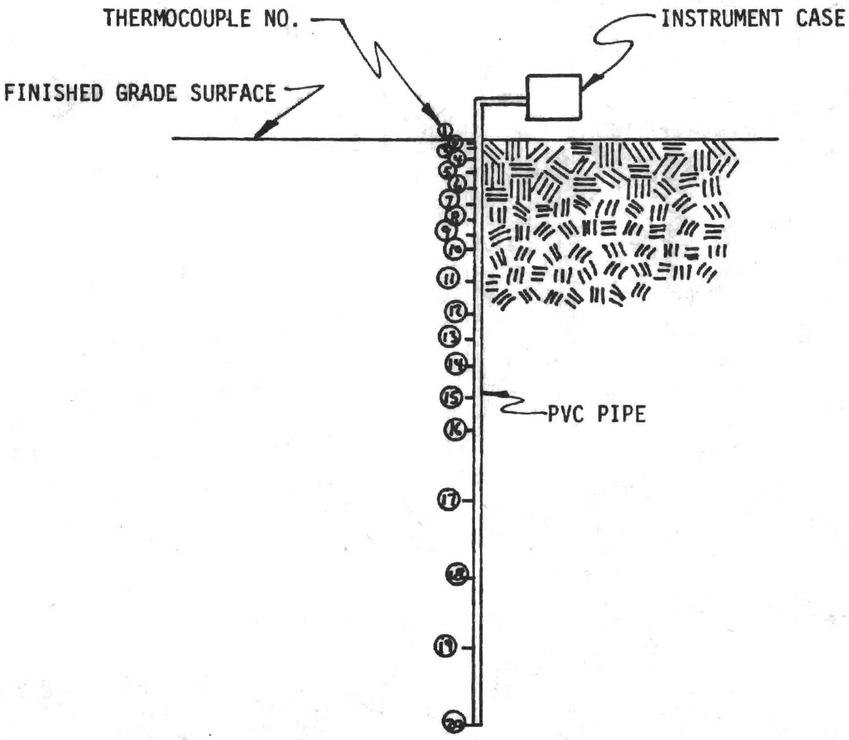

Figure 2 Crossection of Temperature Wells

TABLE I
THERMOCOUPLE DEPTH

THERMOCOUPLE NO.	DEPTH
1	Surface
2	6"
3	1'
4	2'
5	3'
6	4'
7	5
8	6
9	7
10	8
11	10
12	12
13	14
14	16
15	18
16	20
17	25
18	30
19	35
20	40

thermocouple tube has been located vertically in the 40 ft. holes, the holes will be back filled with a 50/50 mixture of Bentonite and Cement. This mixture has been found satisfactory for such applications because it flows readily and should not subsequently develop cracks which might affect soil temperatures.

Soil temperatures will be measured for the 20 locations in each hole at 1 day intervals beginning as soon as the holes have been filled until the thermocouple temperatures begin to stabilize. Once the effect of the installation has disipated, the thermocouples will be monitored once each week. Average daily ambient temperatures will also be measured at the test site so that ambient affects on sub-surface temperatures can be calculated.

While it would be desirable to have some measurement of soil moisture in the vertical wells, lack of suitable soil moisture probes has precluded installation of moisture measuring equipment. It had been anticipated that the moisture probes developed by the U.S. Forest Products Laboratory at Athens, Georgia, would be satisfactory for soil moisture measurements. J.E. Duff of the Forest Products Laboratory advises that these probes will not be satisfactory for soil moisture measurements. After talking to J. E. Duff, Tom Bligh of M.I.T, Phil Metz of BNL, Jim Hartley of Georgia Tech, and Georgia Power, it has been concluded that no satisfactory remote soil mositure measurement device exists within our operation and budget parameters. Dr. Hartley is working with Georgia Power on the development of a probe and expressed interest in evaluating one of their probes in the coil field. If this probe becomes available, it will be installed with the coil field. At present, periodic vertical core samples appear to be the only reliable moisture measurement technique.

Equipment for simulating the thermal load, for pumping the water, for measuring the water flow rate and for measuring the energy input is being identified and ordered so that it will be available for installation with the pipe field. The pipe field configuration will be dependent upon the results from the computer simulations.

Methods for evaluating the effect of pipe diameter, length, spacing and material have also been under investigation during the past two months. Ingersoll[1] gives an exact solution for energy flow from buried tubes which lends itself well to investigating the effect of each of these parameters. Although far from complete, this investigation permits several conclusions at the present time. Figure 3 shows the temperature difference between the soil adjacent to the pipe and soil remote (undisturbed) from the pipe for a constant energy input of 50 Btu/hr/ft as a function of time. Figure 4 shows the temperature differences if the energy per ft^2 of pipe is held constant. When one looks at Figure 3, one might come to the conclusion that larger pipe diameters are desirable. Figure 4 shows this not to be the case. A two inch diameter pipe two feet long has much less temperature differential than a four inch diameter pipe one foot long for the same total energy transferred. This becomes even clearer on Figure 5 where energy transfer rate (Bty/hr. ft^2. ^oF) is plotted versus pipe diameter. Figure 5 shows that smaller pipes have much higher transfer rates.

Basically Figures 4 and 5 show that smaller long pipes are better than larger short pipes. This is not unpredictable if one recognizes that energy flow to the pipe is essentially radial and that small long pipes are exposed to more undisturbed soil than large short pipes.

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[1] Ingersoll, L.R., Zobel, O.J., and Ingersol, A.C., "Heat Conduction with Engineering and Geological Application," McGraw-Hill Book Company, Inc. 1948.

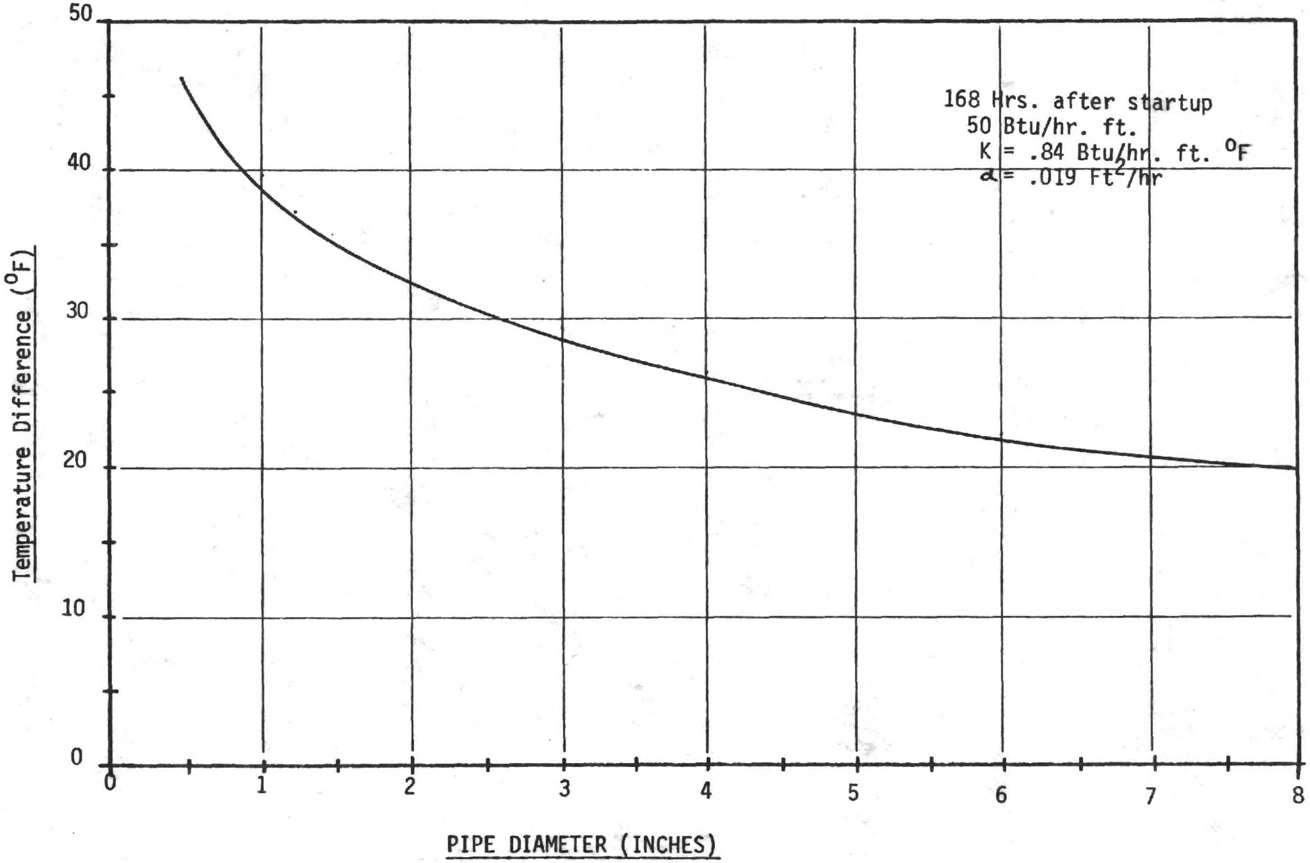

Figure 3 Pipe-Far Field Temperature Difference vs Pipe Diameter (in)

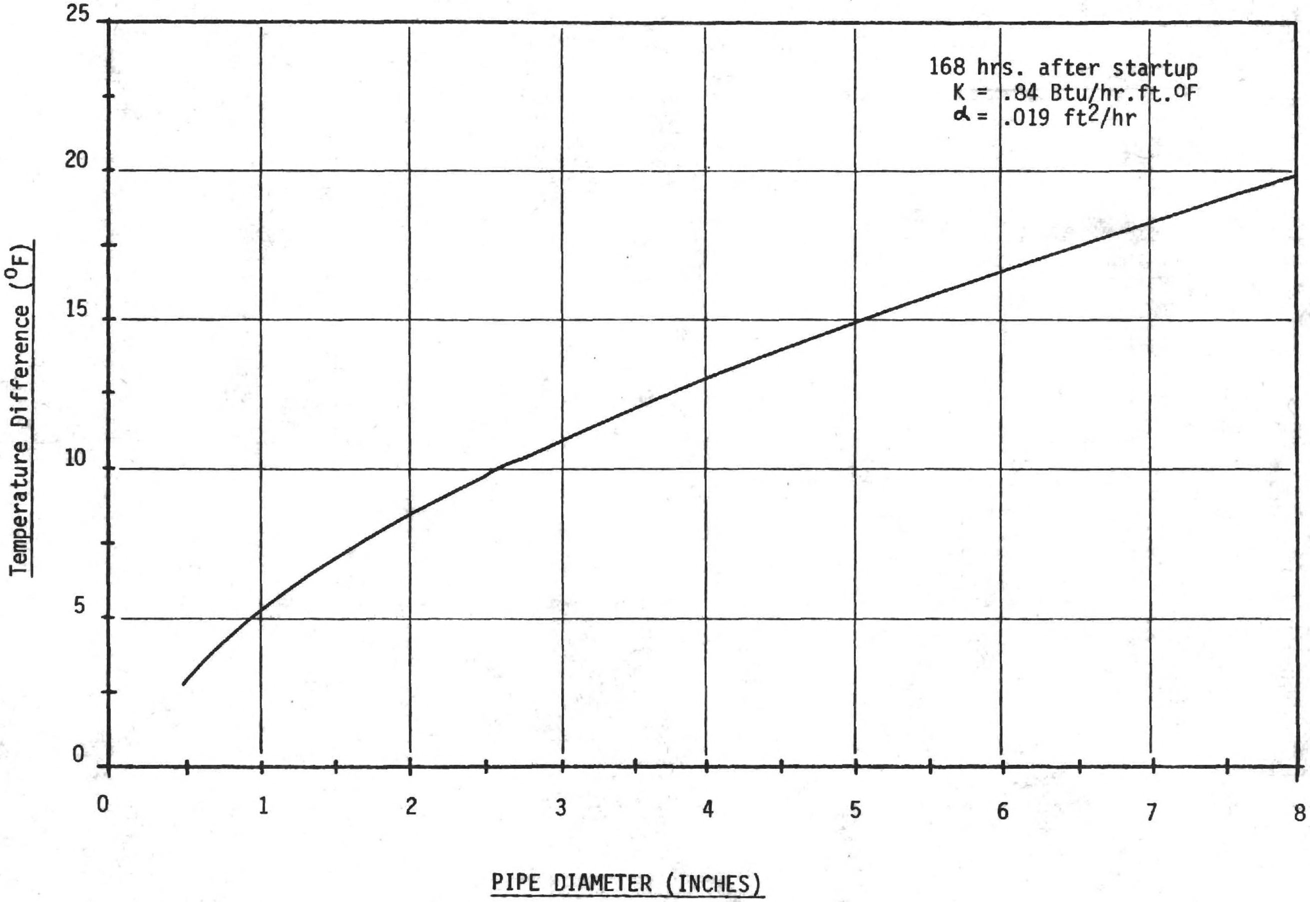

Figure 4 Pipe-Far Field Temperature Difference as Function of Pipe Diameter
(Energy Transfer Per Ft^2 of Pipe Held Constant)

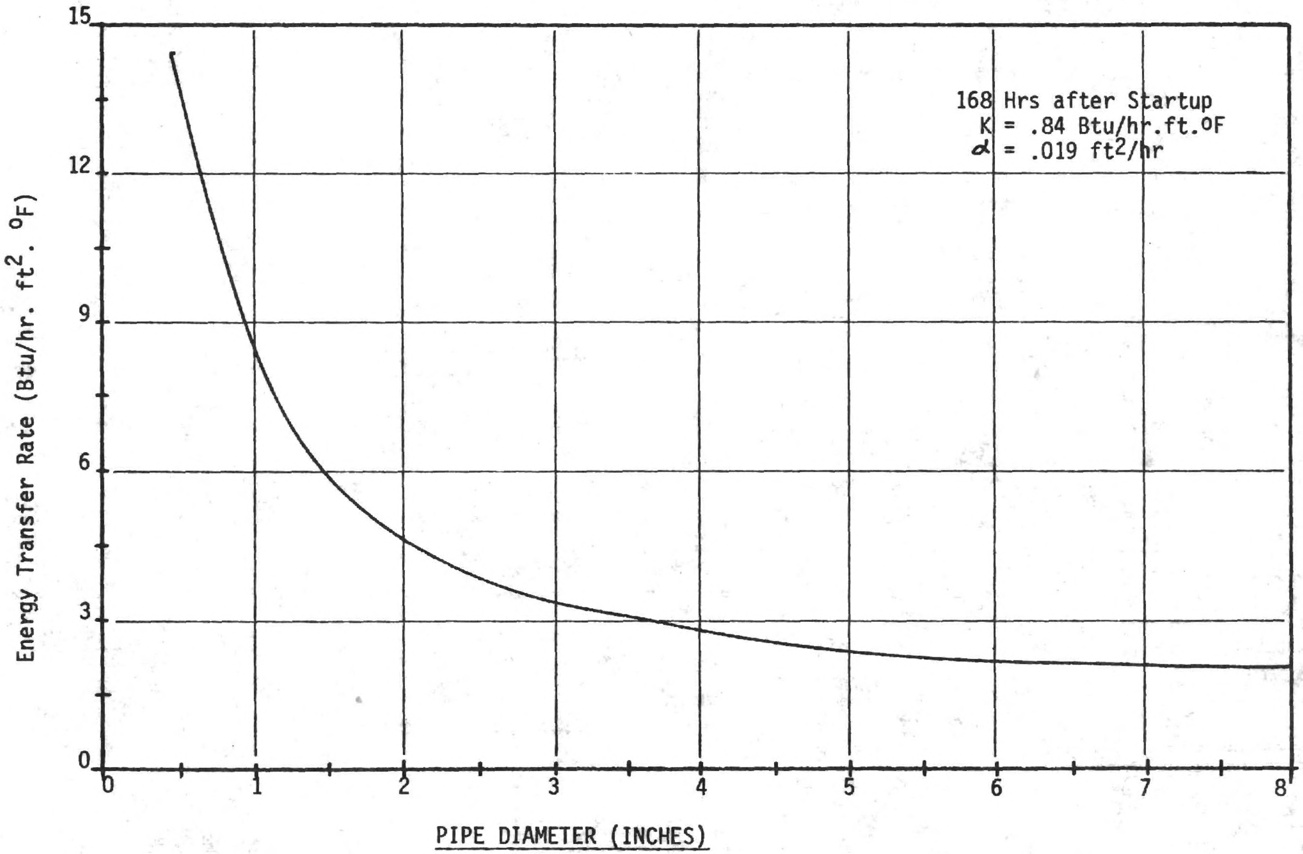

Figure 5 Energy Transfer Rate as a Function of Pipe Diameter

While the calculations show that the smaller the pipe the better its operation from an energy transfer standpoint, there are other factors, such as pressure drop through the pipe, trenching costs and space available, which will influence pipe diameter selection.

Figure 6 shows the effect of time on energy transfer rate. This figure shows the energy transfer rates to drop quite rapidly during the first 50 hours (2 days), drop less rapidly during the next 100 hours (4 days) and essentially become constant after 168 hours (7 days).

Figure 7 shows the effect of soil thermal conductivity and diffusivity on temperature difference. Comparison of Figure 3 and 7 shows that lower thermal conductivity and diffusivity significantly increase the temperature difference required for a given heat transfer rate. It is believed the values used to calculate the data used to construct Figure 3 are more nearly representative of the soil in Atlanta. As new data becomes available from out tests, we will refine these calculations.

Figure 8 shows the effect of energy withdrawal rate on temperature difference. As one would expect higher withdrawal rates require higher temperature differentials. The equations used to develope these curves have been programmed on a programmable calculator and are being changed to let temperature difference determine transfer rates rather than rates determine temperature difference. The program is capable of superimposing the effect of temperature waves from the surface and from adjacent pipes on the energy transfer rate.

Work during the next month will concentrate on getting the vertical holes installed and instrumented and optimizing the pipe size, material and layout. We are on schedule and do not see problems which might cause delays.

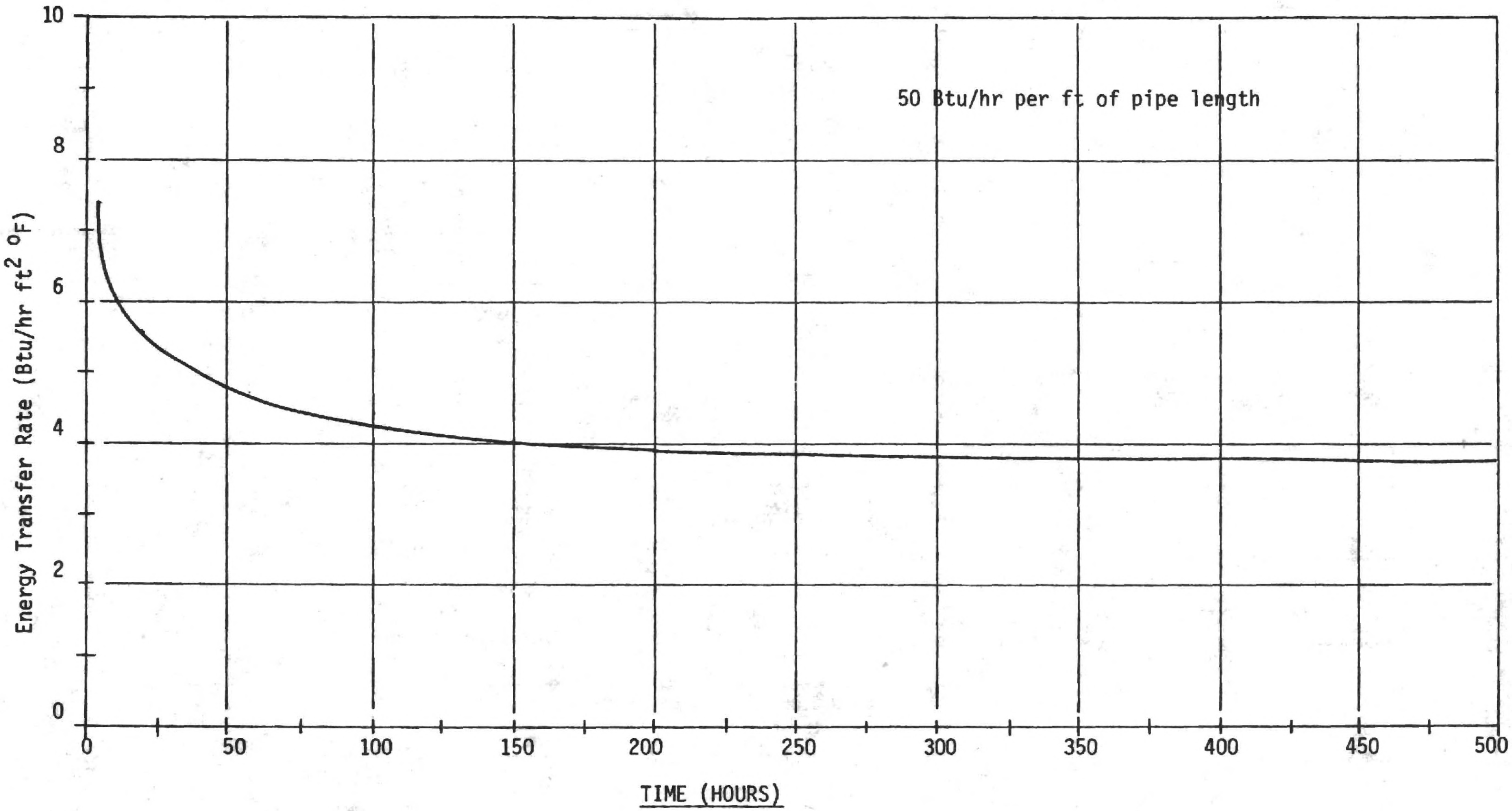

Figure 6 Energy Transfer Rate as a Function of Operating Time

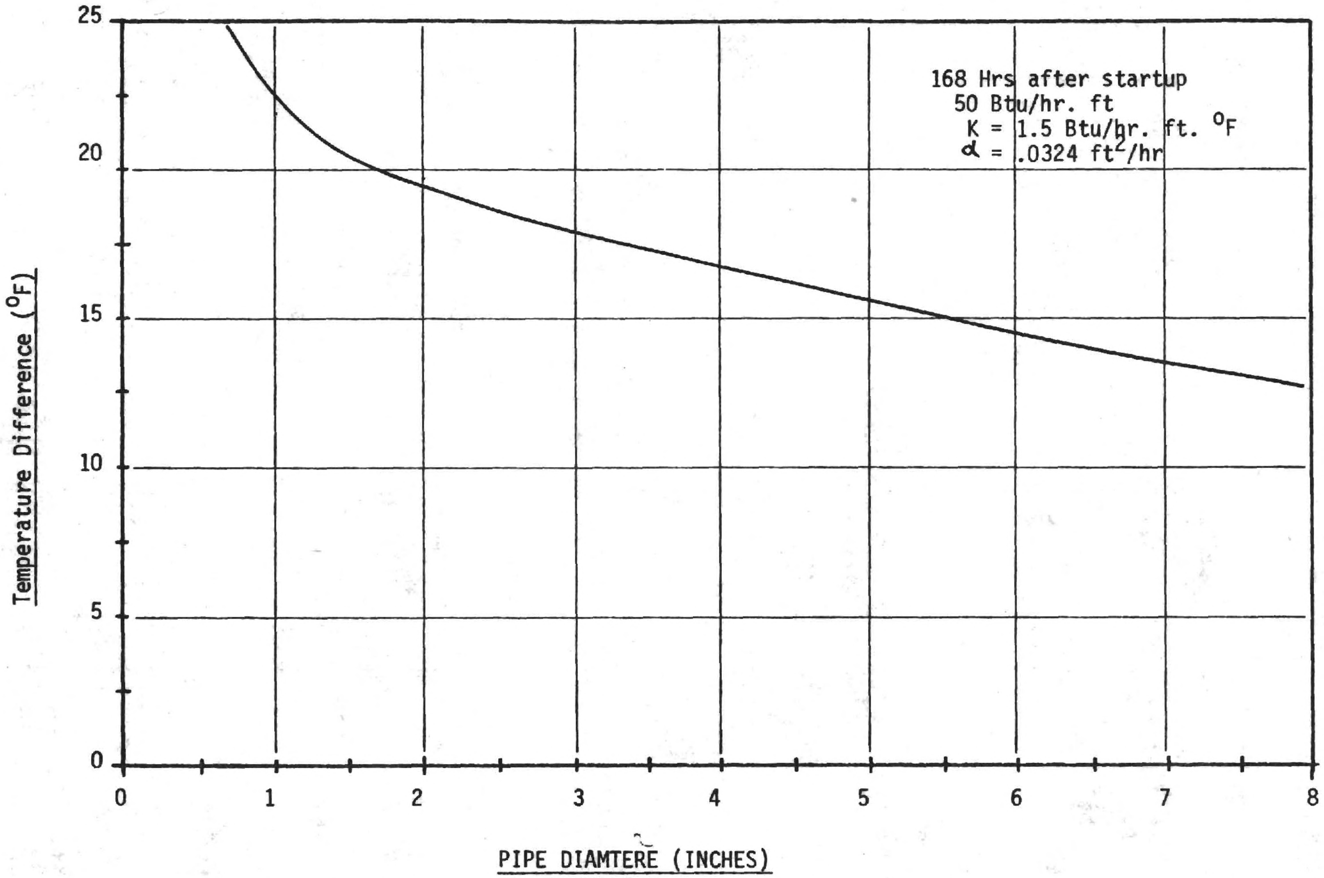

Figure 7 Pipe-Far Field Temperature Difference vs Pipe Diameter (in)

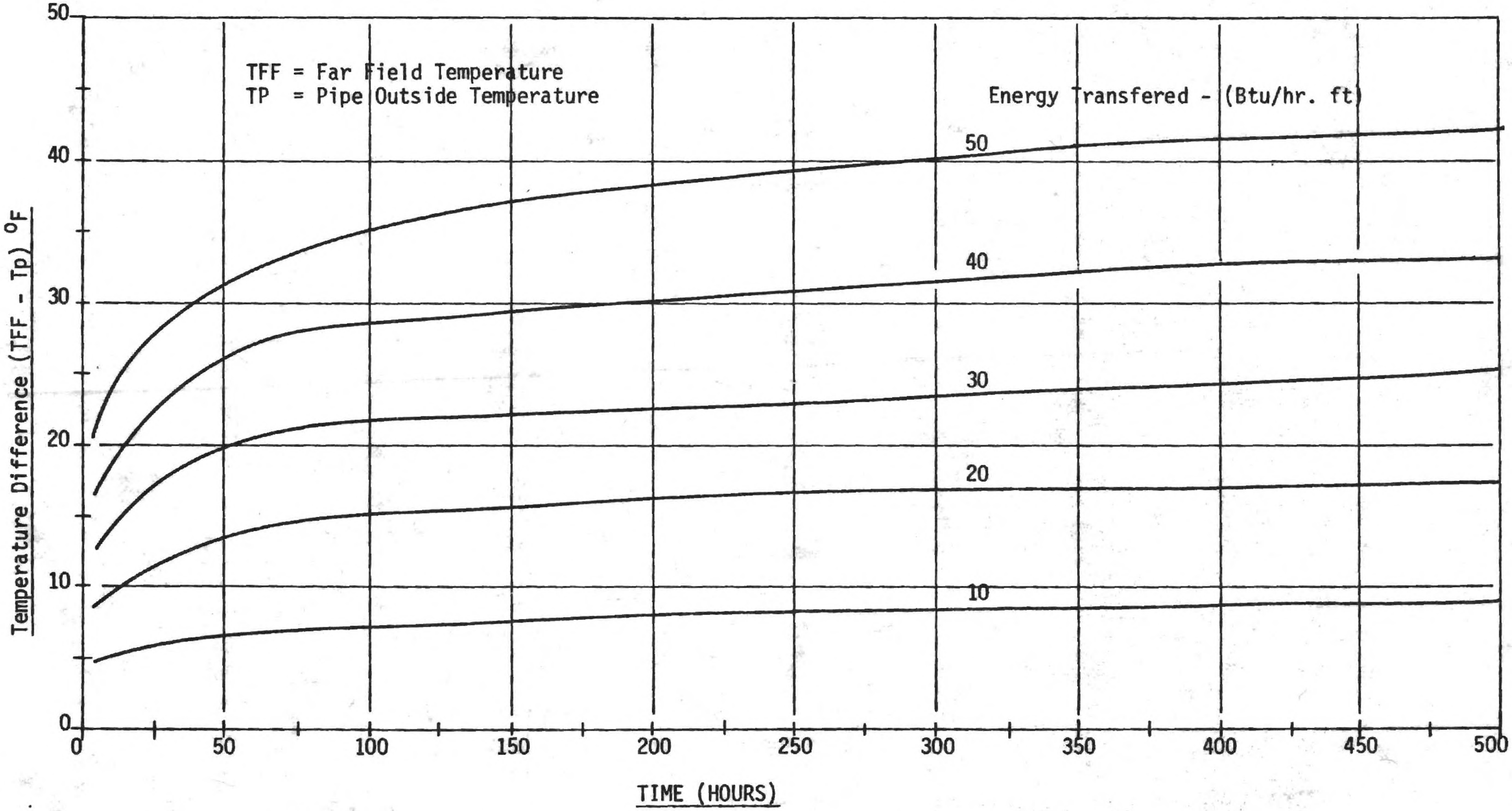

Figure 8 Pipe - Far Field Temperature Difference vs Operating Time

D-48-636

INVESTIGATION OF PASSIVE COOLING
FOR HOT-HUMID CLIMATES

Progess Report
No. 4

Submitted to the

Research and Development Branch
Heating and Cooling, Conservation and Solar Applications
Department of Energy

By The

College of Architecture
Georgia Institute of Technology
Atlanta, Georgia 30332

June 16, 1980

Project Director

JAMES M. AKRIDGE
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INVESTIGATION OF PASSIVE COOLING FOR HOT-HUMID CLIMATES

Work this month has been directed toward the installation of the two forty foot vertical wells for soil temperature measurements, the investigations of methods for installation of field ground temperature probes, and the investigation of field layout using the BNL simulation routines developed for TRNSYS.

The two forty foot soil temperature measurement wells were sunk as explained in report number 3. Well number 1 was sunk as planned while well number 2 was moved to the position shown in figure 1. Well 2 was moved because of a concern that the insulated steam line might influence the temperature reading at the planned location.

Surprisingly, both wells hit water at 20-23 feet. This was surprising because all the soils specialists in the area said we wouldn't expect to hit ground water anywhere in the Atlanta area. The particular site chosen is an old stream bed with a mixture of silt and clay as the predominant toil type. Apparently, considerable water remains at depths greater than twenty feet despite the storm sewer shown on figure 1.

Soil samples were taken every five feet during the drilling operation. These samples were weighed, moisture content determined and preliminary conductivity, diffusivity and heat capacity measurements made. Due to the preliminary nature of the soil property measurements they will be further refined and included in the next progress report.

Initial soil temperature measurements in hole #1 are very close to those predicted from the computer runs. Soil temperatures measured in hole #2 are several degrees lower than expected. It is not certain, at this time, whether the low reading from hole #2 results from its proximity to a large tree, whether the readings are in error or whether they are correct and those measured in hole #1 are in error. Temperatures in the two holes at depths below the water level are identical. It is expected that these questions will be resolved within the next month.

Subsequent to the installation of the two forty foot holes, experiments with a water drill have demonstrated the feasibility of drilling holes quite rapidly. Several additional soil temperature holes will be drilled to the twenty foot depth using the water drill. Instrumentation of these additional holes should resolve the questions we now have about the temperatures measured in the first two holes.

Simulation of different fields using the BNL field simulation program has progressed much less rapidly than anticipated due to lack of familiarity with the program and the considerable data processing necessary before the program can be run. Sufficient work has been accomplished to feel confident about the four foot

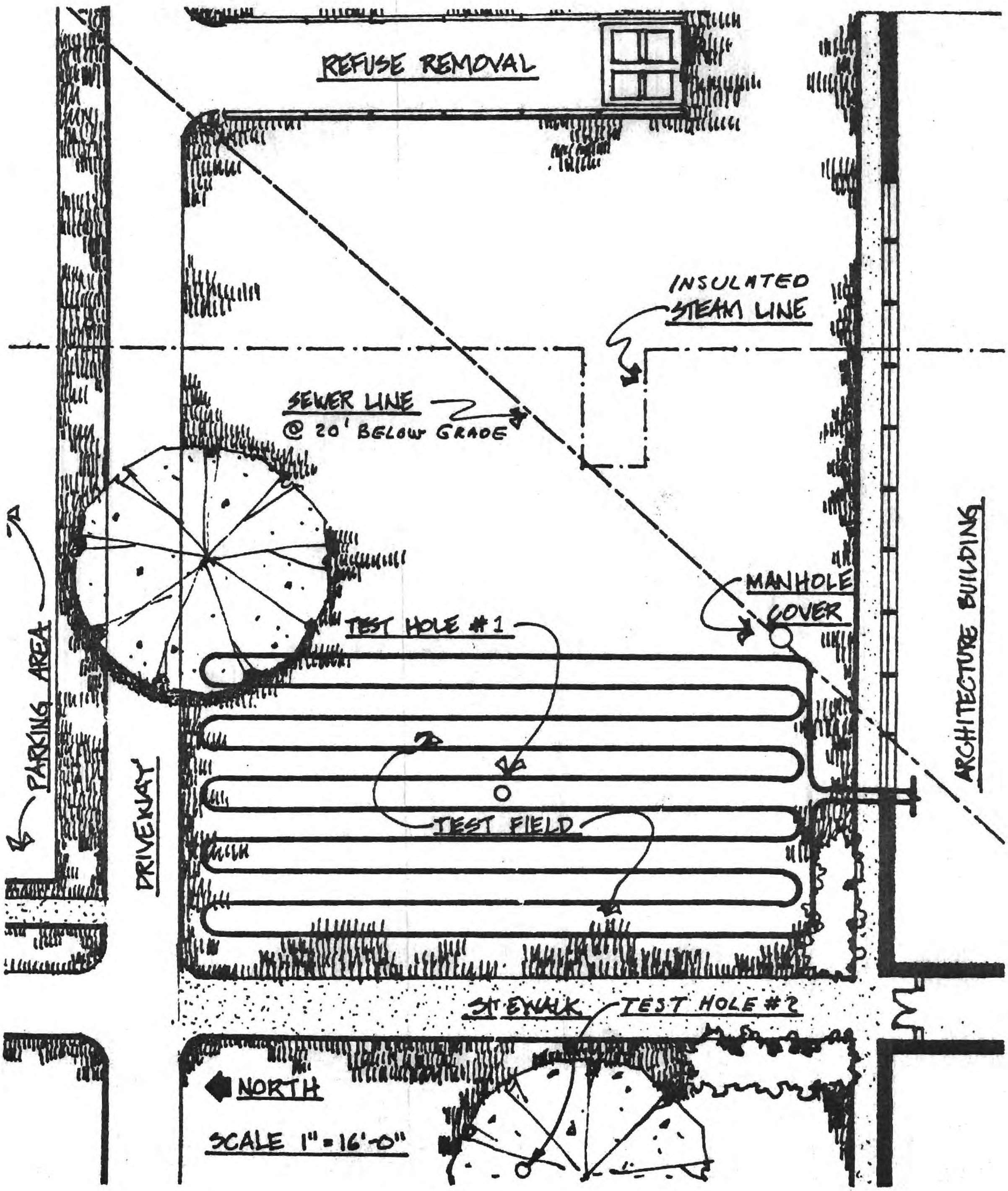

FIGURE NO. 1 Plan of Test Site

spacing tentatively chosen for the experimental field. Work in this area will intensify during the next several weeks and the results reported in the next progress report.

Work during the next month will concentrate on reolving the questions regarding the ground temperatures, optimization of the experimental field and initiation of the field installation. Work has also just begun on the MITAS simulation work. This work will aloso be accelerated during the coming weeks. Soil property measurements should also be completed within the next month.

D-48-636

INVESTIGATION OF PASSIVE COOLING FOR HOT-HUMID CLIMATES

Progress Report
No. 5

Submitted to the

Research and Development Branch
Heating and Cooling, Conservation and Solar Applications
Department of Energy

College of Architecture
Georgia Institute of Technology
Atlanta, Georgia 30332

July 30, 1980

Project Director

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INVESTIGATION OF PASSIVE COOLING FOR HOT-HUMID CLIMATES

Work during this month concentrated on investigation of field performance using three separate computer simulation programs. The TI 59 "Line Source" program discussed in progress report number 3 has been refined to permit evaluation of the effect of changes in various parameters. "Line Source" is used to denote a solution which looks at the energy source or sink as a line as opposed to a point or plane. This program will evaluate the effect of time, material thickness, material conductivity, soil conductivity, soil diffusivity, mass flow rate, fluid specific heat, and length of pipe. A second simulation program, the GROCS program, was obtained from Brookhaven National Laboratories and modified to work with the 10.1 version of TRNSYS. Simulation using this model has permitted the simulation of the exact field which will be used in our experiments. Evaluation of different parameter, such as depth of pipe, pipe spacing, insulation thickness, and total pipe length is possible with GROCS. Although simulation will continue, we have arrived at a field which we feel is satisfactory for experimental purposes, and is close to what one would use in an actual installation. A third program, MITAS, obtained from the Martin-Marietta Corporation and used in many other research programs at Georgia Tech and the Engineering Experiment Station has been used to evaluate the effect of pipe spacing, pipe depth, insulation thickness, and total block size. This program which is a multi-node simulation routine requires extensive computer time and is therefore not as easily used or as well suited to the optimization process as the previous two programs. It is considered to be more accurate. Simulation of the 750 ft. long field presently being considered for installation at Georgia Tech required about an hour and a half of computer time and over 1800 nodes for simulating a 3000 hour run using the MITAS model. The three simulation programs, when given the same boundary conditions, agree amazingly well. Figure 1 shows the results of simulations assuming an infinite isothermal boundary condition, that is no limits on the block size, for two of the simulation programs. One will notice that the line source program and the GROCS program obtained from BNL never deviate more than 1/2 a degree throughout the 2,000 hours of the simulation. The MITAS program also agrees very closely with the other two. We now feel confident that the three programs are capable of providing the simulation we need for optimizing the field. Each program will be used, and is being used in the area in which it is best suited. Results from a number of parametric studies will be given in later sections of this report.

Work this month has also concentrated on the installation of a third vertical probe for measuring earth temperatures and the continued monitoring of earth temperatures in the two 40 ft. holes installed previously. Results of these measurements are given in a separate section.

The "Line Source" program discussed in progress report No. 3 and previously executed on the TI 59 has been found to be much better suited for sensitivity analysis and parametric studies than the other two programs. As a result, the program has now been transferred to the CDC 70/74 which permits a much more rapid evaluation of the effect of various parameters.

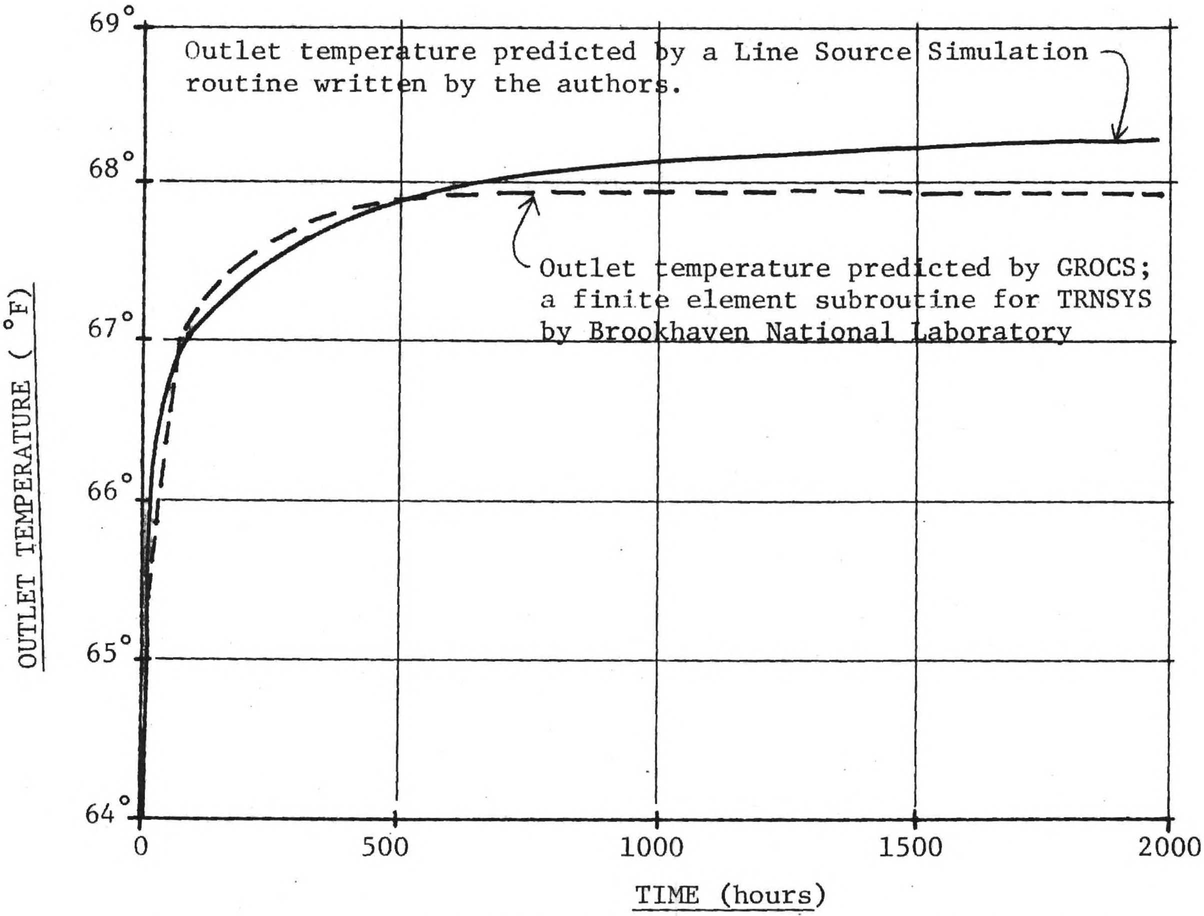
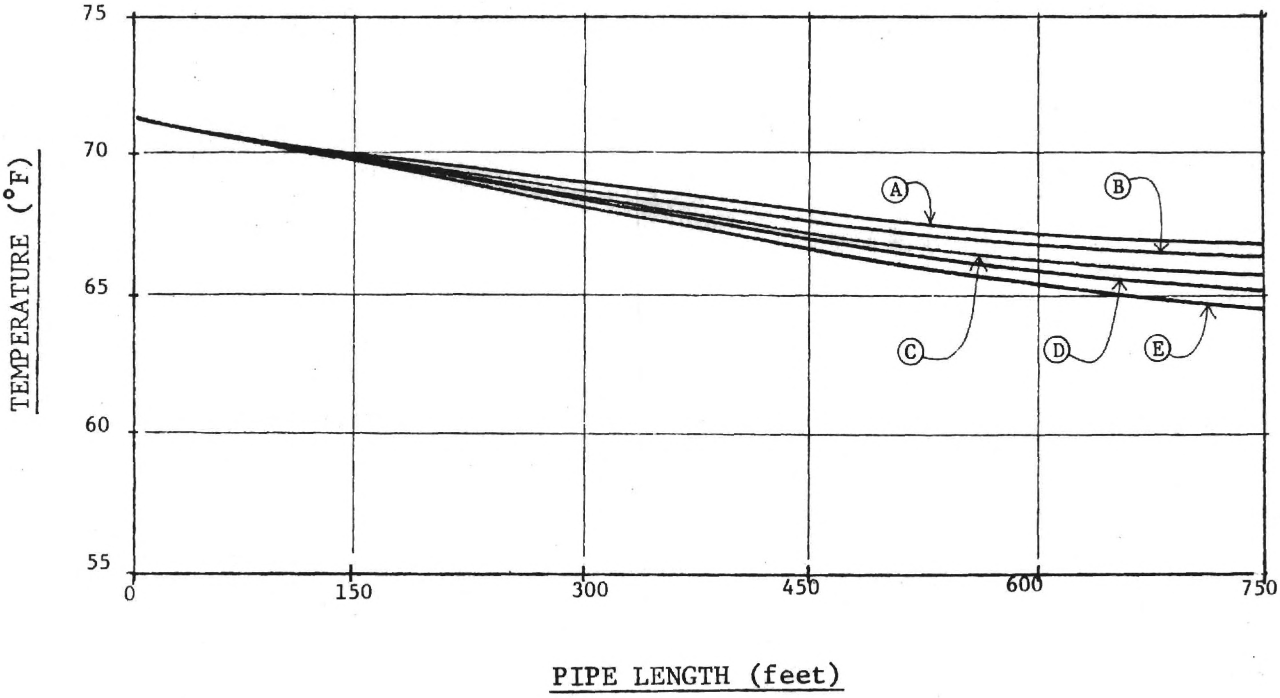


FIG. NO. 1 A Comparison of Earth Coupled Modelling Technique

OUTLET TEMPERATURE vs TIME is plotted for a 1½" DIA. polyethelene water pipe buried @ 4'0" below grade. Mass flow through the system is 2000 lbs. of water per hour at 72°F inlet temperature. Earth temperature at start up is an uniform 55°F.

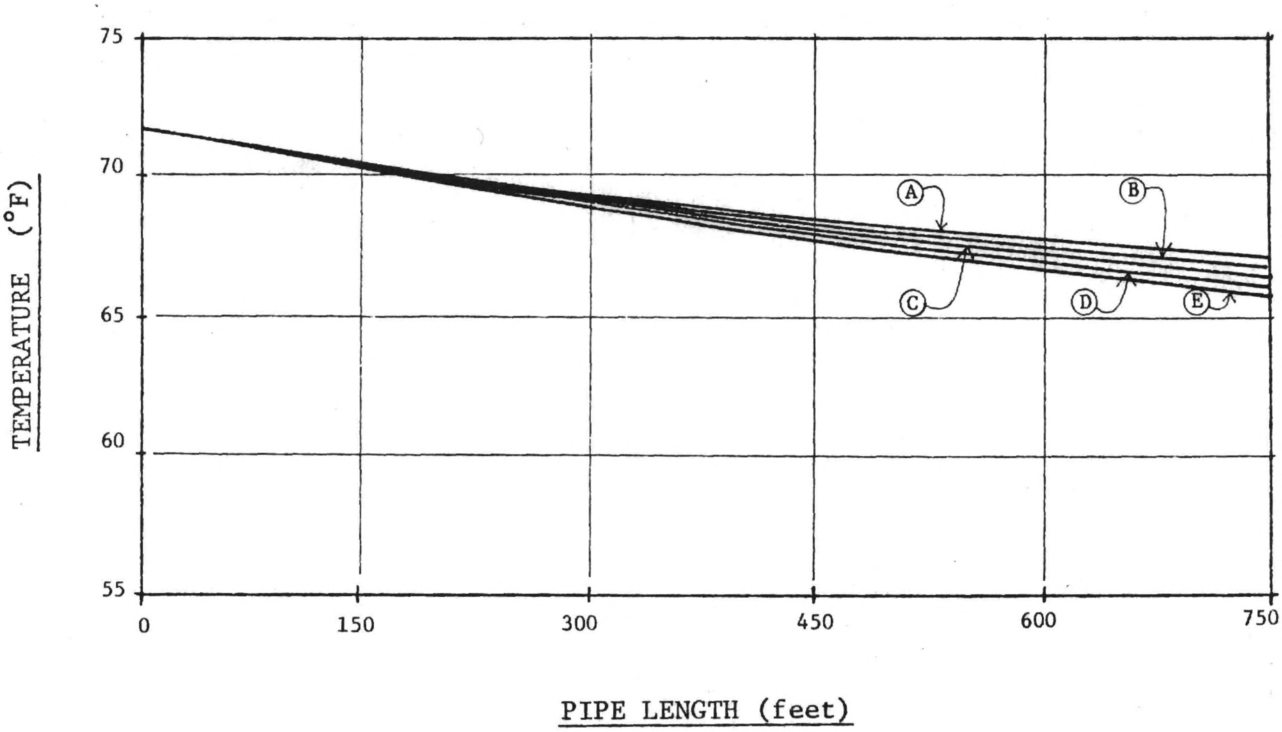
Figure 2 expands on the work reported on in progress report No. 3 where the effect of pipe diameters on energy transfer rate was evaluated. Figure 2 shows that smaller pipes have energy transfer rates almost as high as larger pipes for a given length. If pipe area is kept constant the smaller pipes have much higher transfer rates. Since smaller diameter pipes are much easier to position and much less costly to purchase and install and will bend into configurations more nearly suited to what is needed in this study, we have arrived at a nominal pipe diameter of 1 1/2 inches. Since polyethene pipe is the least expensive pipe available which is suitable for this purpose, we have settled on the use of low density polyethene pipe 1 1/2 inch nominal inside diameter for further studies. This type and size is the same chosen by Brookhaven National Laboratories in their earth coupled heat pump studies. Figure 3 shows the effect of material conductivity while figure 4 shows the effect of material thickness. One will notice that there is very little difference in the energy transfer rate between an aluminum pipe with a very high conductivity and that of the plastic pipe with the lowest thermal conductivity. One will also notice that thickness has very little effect on energy transfer rate.

Figure 5 shows the effect of soil diffusivity on energy transfer rate or temperature along the length of pipe. Again for an infinite source, soil diffusivity has little effect on the energy transfer rate. It does become significant for realistic block sizes. This is still being evaluated. Figure 6 shows that mass flow rate has a significant impact on the energy transfer rate along the pipe. At the lower flow rate the fluid quickly approaches the soil temperature, thus greatly reducing transfer rate.



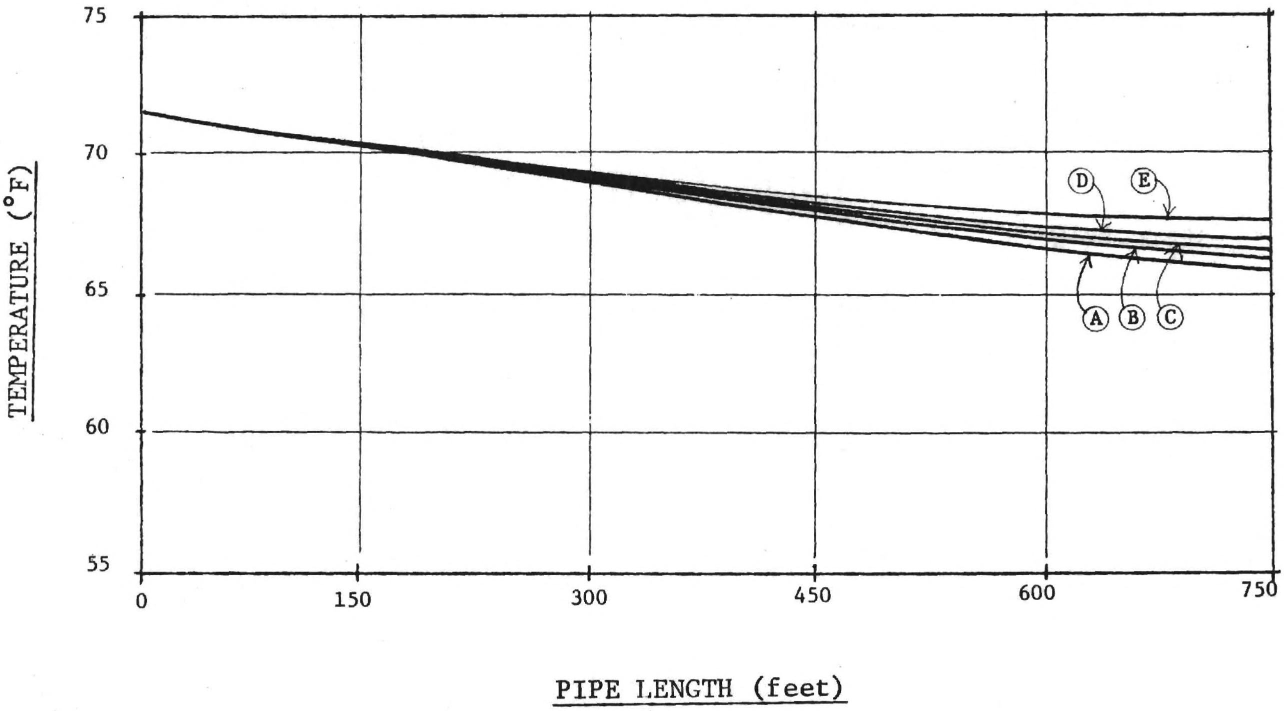
PIPE RADIUS		HEAT TRANSFER AT END OF RUN
CURVE A = .5 IN		TOTAL Q = 9,819 BTU/HR
" " B = .75 IN		" " " = 10,837 BTU/HR
" " C = 1.0 IN		" " " = 11,597 BTU/HR
" " D = 1.5 IN		" " " = 12,749 BTU/HR
" " E = 2.0 IN		" " " = 13,644 BTU/HR

FIG. NO. 2 TEMPERATURE vs PIPE LENGTH for several values of PIPE RADIUS



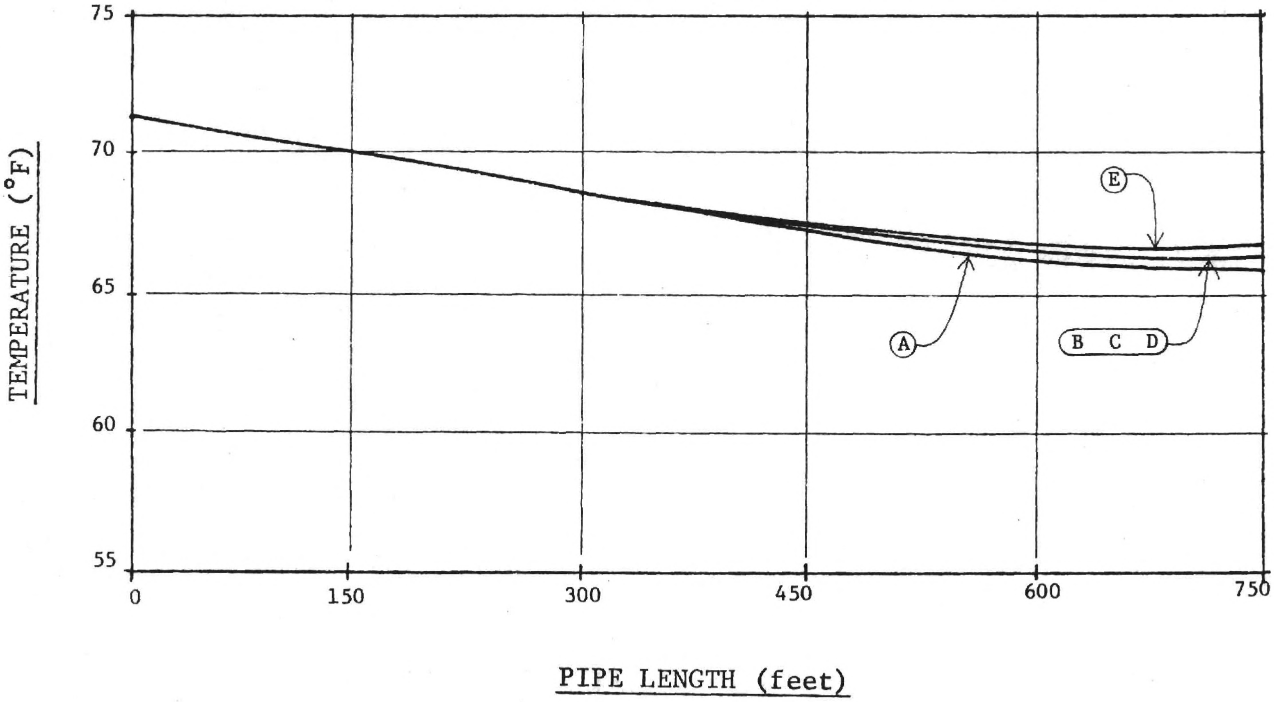
TYPES OF PIPE		HEAT TRANSFER AT END OF RUN
CURVE A = POLYPROPYLENE		TOTAL Q = 9,630 BTU/HR
" " B = PVC		" " " = 9,760 BTU/HR
" " C = ABS		" " " = 10,392 BTU/HR
" " D = POLYETHELENE		" " " = 11,025 BTU/HR
" " E = COPPER		" " " = 11,955 BTU/HR

FIG. NO. 3 TEMPERATURE vs PIPE LENGTH for several types of PIPE



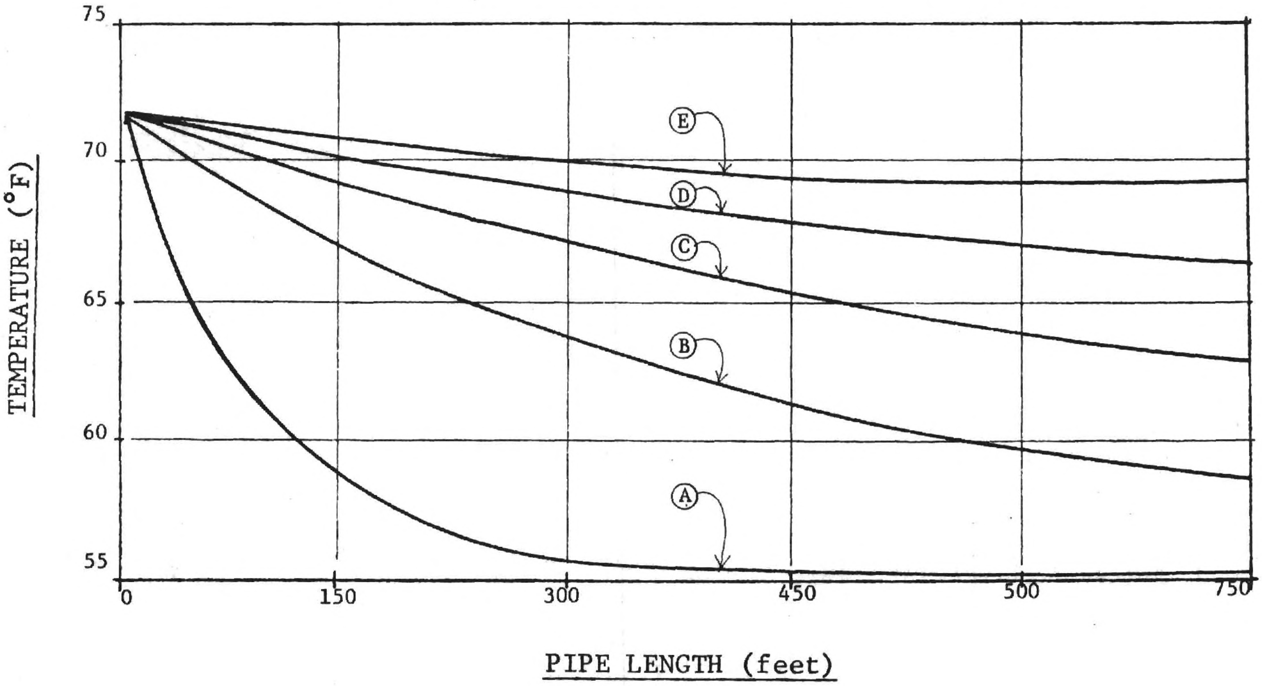
WALL THICKNESS	HEAT TRANSFER AT END OF RUN
CURVE A = .094 IN	TOTAL Q = 11,144 BTU/HR
" " B = .125 IN	" " " = 10,900 BTU/HR
" " C = .188 IN	" " " = 10,434 BTU/HR
" " D = .25 IN	" " " = 10,013 BTU/HR
" " E = .5 IN	" " " = 8,608 BTU/HR

FIG. NO. 4 TEMPERATURE vs PIPE LENGTH for several values of WALL THICKNESS



SOIL DIFFUSIVITY	HEAT TRANSFER AT END OF RUN
CURVE A = .02 SF/HR	TOTAL Q = 11,625 BTU/HR
" " B = .025 SF/HR	" " " = 11,403 BTU/HR
" " C = .03 SF/HR	" " " = 11,228 BTU/HR
" " D = .035 SF/HR	" " " = 11,084 BTU/HR
" " E = .04 SF/HR	" " " = 10,962 BTU/HR

FIG. NO. 5 TEMPERATURE vs PIPE LENGTH for several values of SOIL DIFFUSIVITY



MASS FLOW RATES		HEAT TRANSFER AT END OF RUN
CURVE A =	100 LBS/HR	TOTAL Q = 1,699 BTU/HR
" " B =	500 LBS/HR	" " " = 6,766 BTU/HR
" " C =	1000 LBS/HR	" " " = 9,319 BTU/HR
" " D =	2000 LBS/HR	" " " = 11,144 BTU/HR
" " E =	5000 LBS/HR	" " " = 12,483 BTU/HR

FIG. NO. 6 TEMPERATURE vs PIPE LENGTH for several values of MASS FLOW

As will be discussed later, block capacity may be the limiting factor rather than transfer rate, thus lower flow rates may be desirable.

Since the "Line Source" program was developed primarily to examine heat transfer in a linear field with isothermal boundary conditions and for sensitivity analysis under these conditions, it is not well suited for evaluating the effect of pipe spacing, block size, insulation location or thickness. As a result of these limitations, the GROCS program obtained from BNL has been used extensively to evaluate the effect of pipe spacing, pipe depth, insulation thickness insulation location and block size. GROCS provides a realistic boundary condition and allows the analysis of the impact of the ambient temperature pulse on the earth heat sink. GROCS is providing an indication that the absolute capacity of the earth block is a limiting factor early in the cooling season, while the effect of the ambient pulse dominates during the latter season. In a real situation one is limited in block (field) size, either due to the installation cost or space available for a block. Figure 7 shows the field layout presently being proposed for experimental purposes. We feel this block is realistic in size and our simulation routine has been primarily directed towards determining the field layout which will permit extraction of the greatest amount of energy from the block throughout the cooling period and the storage of the greatest amount of cooling capability in the block during the winter charging period. These simulations have shown that a large portion of the cooling storage capability is potentially lost if the field lacks adequate insulation.

We are currently examining the possibility of using a TRNSYS model, including an air to water heat exchanger routine to examine the winter

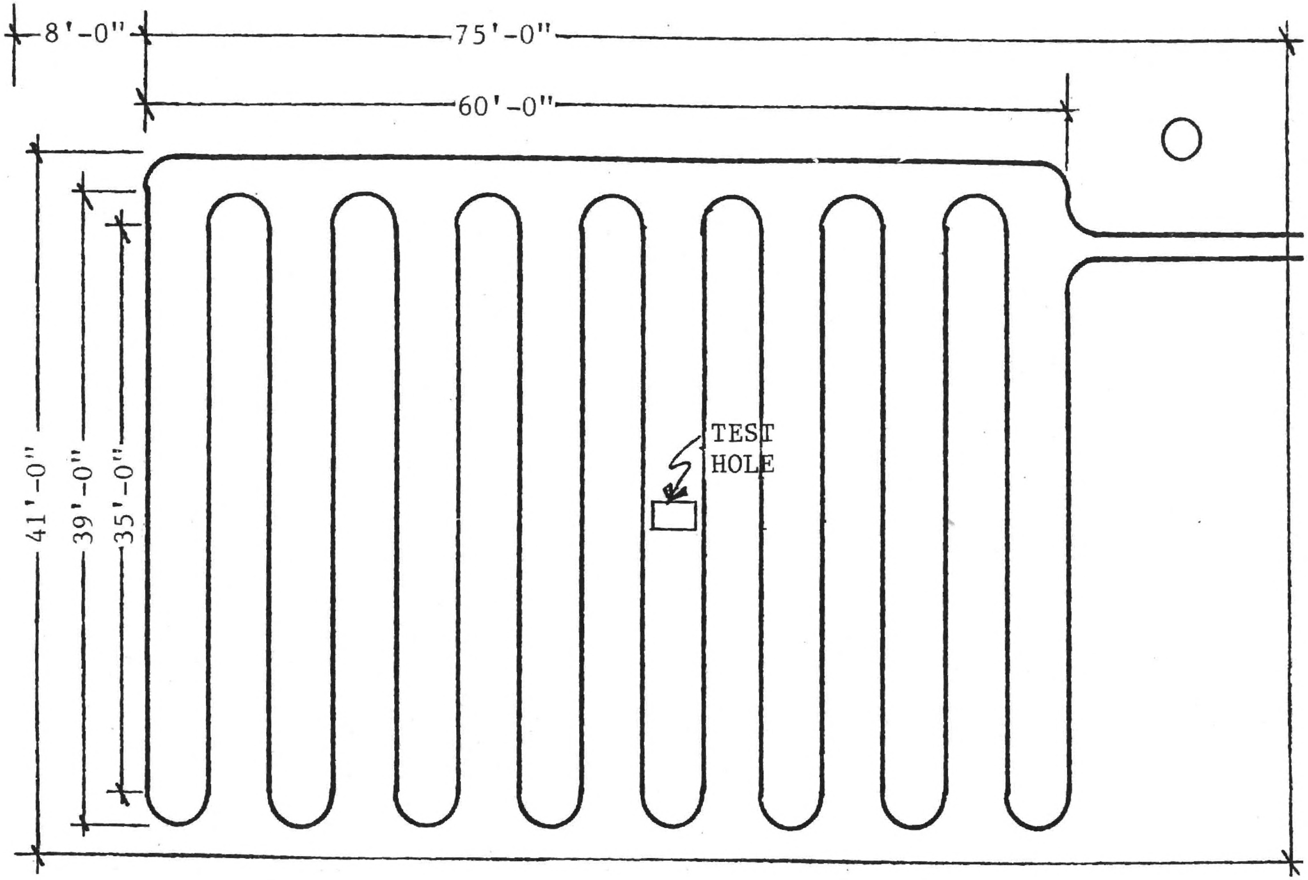


FIG. NO. 7 PROPOSED PIPE FIELD LAYOUT

season cooling of the earth field using ambient temperature as a driving force. The combination of GROCS and TRNSYS will allow us to arrive at temperature distributions throughout the block using realistic weather data, rather than an assumed block temperature or has been done during the preliminary optimization studies. This charging simulation routine has just begun although it should be completed by the end of the next report period.

The GROCS model has permitted us to determine that ambient temperature pulses dominate the late season cooling, even with R8 insulation. This indicates the desirability of either locating the field underneath a structure or the use of very high insulation values above the field. The GROCS model has also indicated that extraction rate may not be the only limiting factor for total seasonal cooling capacity. Another limiting factor in passive cooling using this concept may be the total block cooling capacity. Very high extraction rates are possible with the 750 ft. length coil. With high extraction rates early in the summer one finds the cooling capacity of the earth block is exhausted before the end of the cooling season.

Initial simulation routines have all assumed that energy will be extracted continously at some given rate, and determined how long this rate could be carried on before the field energy would either be exhausted or the rate could no longer be maintained. The TRNSYS program is now being used to simulate an energy efficient, well insulated house using actual Atlanta weather data, with specific emphasis on the sensible cooling load one would expect in such a house. Latent loads will also be determined with this program. It is expected that by the end of the next report period this model will have been completed; providing an hour by hour cooling load profile for a full year in Atlanta. Once this has been determined, we will be able to use this to determine what the extraction rate from the cooling block is, what the charging rate throughout the charging period is and what the temperatures will be throughout the block at any time of the year. This will allow us to determine whether passive cooling will be able to carry 100% of the sensible cooling or perhaps a lesser percentage as is current practice with passively heated houses. This program will also allow us to determine whether the use of a heat pump during the later stages of the cooling year will improve system performance and also improve the block energy storage capability. One would expect this to be so, but the control strategy needs to be worked out before one can establish exactly how it should be accomplished.

A third vertical hole has been drilled and instrumented with thermocouples to determine ground temperatures. This hole was felt necessary because of the lack of correlation between ground temperatures measured in the two previous holes. The third hole agrees much more closely with the one presently planned to be located in the center of the test field.

Figure 8 and 9 show temperatures measured at various depths for the last three weeks. These temperatures vary about as one would predict using the wave propagation equation down to about 15 ft. Temperatures at the 20-25 ft. level vary much more widely than one would predict. At the present time we feel that this is due to the movement of water through this area which was an old stream bed at one time. There is presently a storm

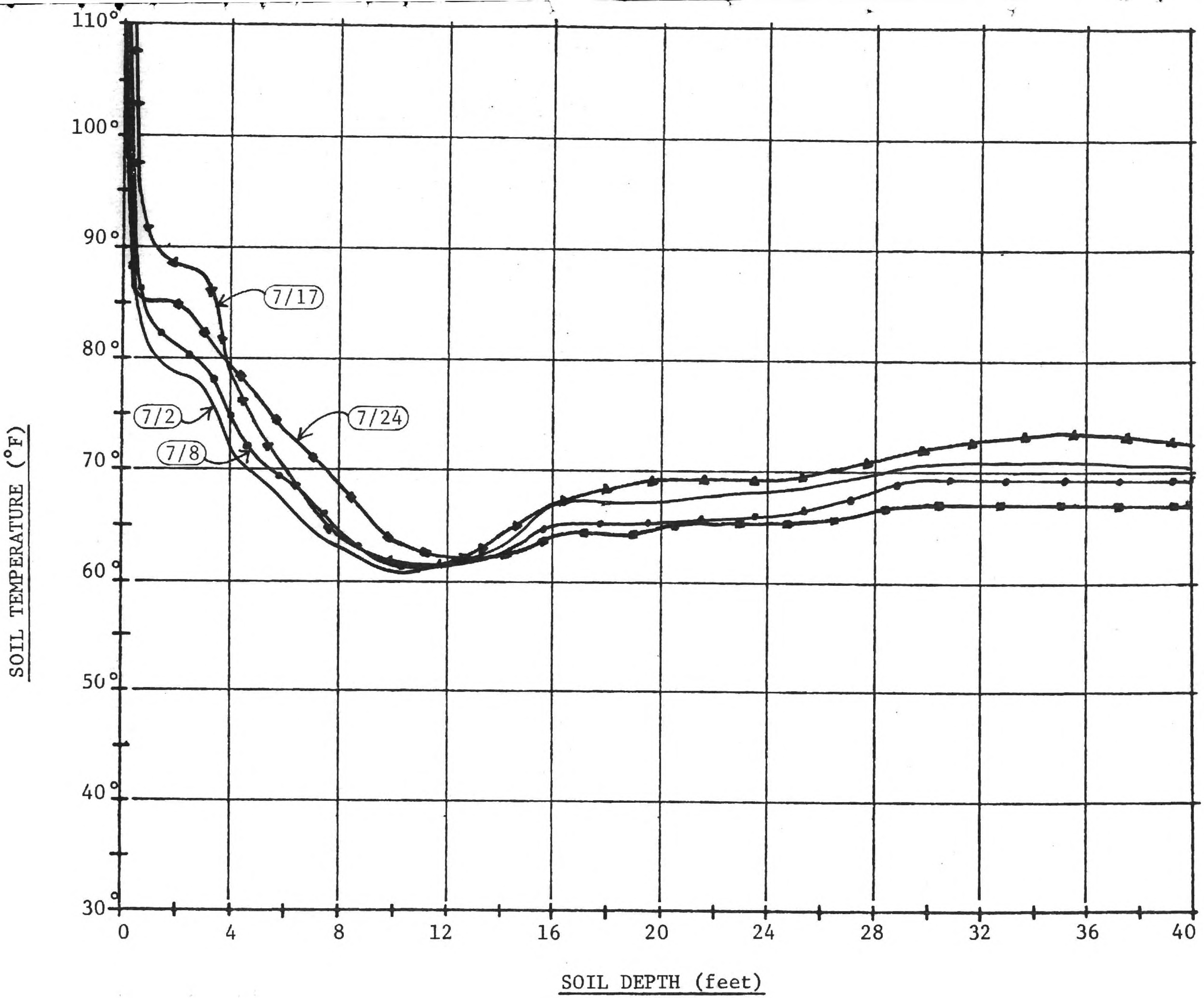


FIG. NO. 8 PASSIVE COOLING STUDY

Weekly Soil Temperature Analysis - Test Hole No. 1

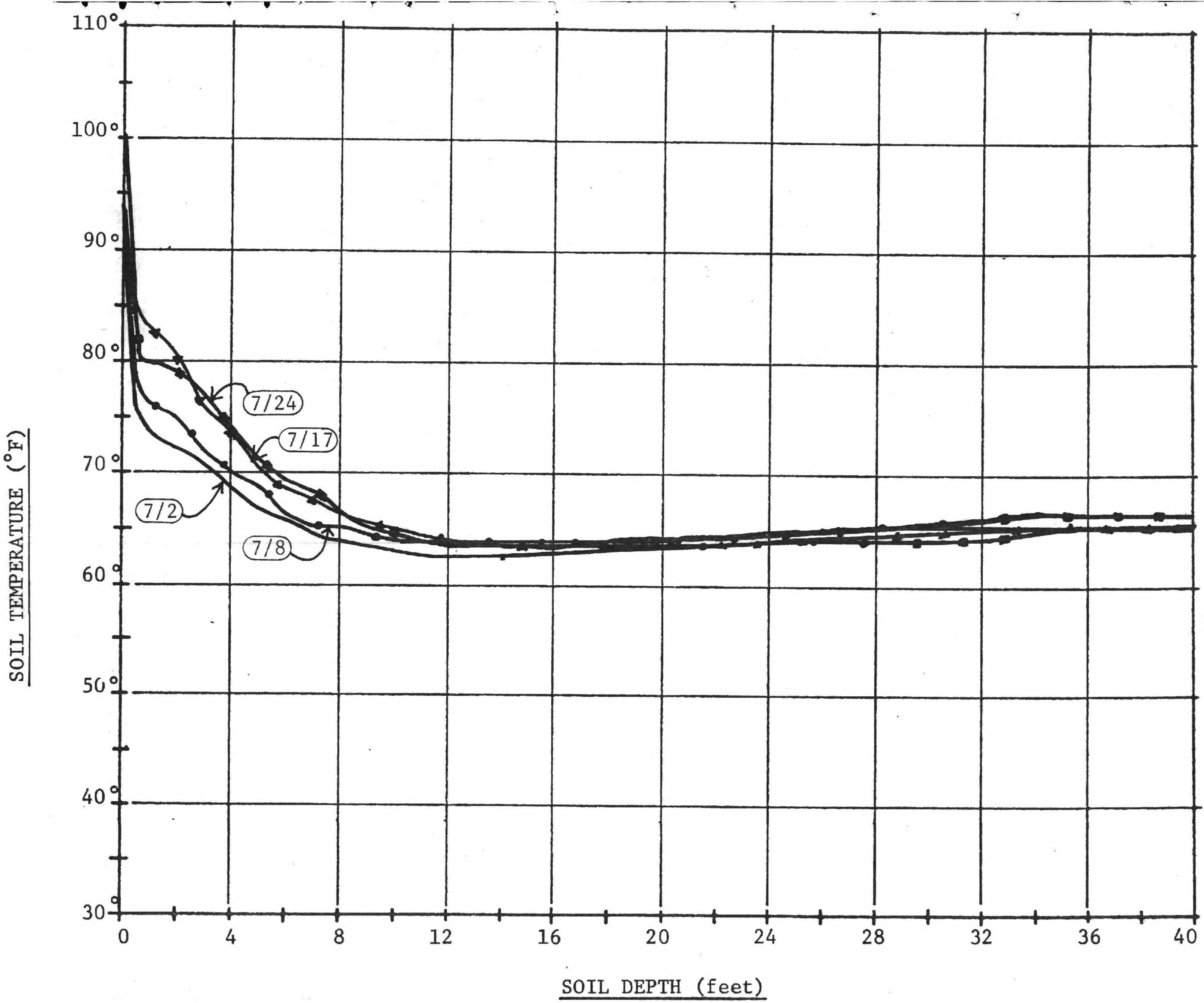


FIG. NO. 9 PASSIVE COOLING STUDY

Weekly Soil Temperature Analysis - Test Hole No. 2

sewer running through the east portion of the test site at a depth of 20 plus feet, which may contribute to the wider fluctuations at the greater depths in hole number one. Temperatures in the reference hole, which is located adjacent to a large tree, do not vary nearly as widely at the greater depths. This is what one would have predicted. We have had instrumentation difficulties with all three holes with two different types of thermocouple readout instruments failing. The thermocouples for test holes one and two currently terminate at the top of the wells. The cause of some of our instrumentation difficulty could be due to the necessity of taking readings for these holes with the instrument outside (130 F at the surface). After installation of the field, the thermocouples will be wired to a permanent reading location indoors, allowing the use of a more sophisticated thermocouple measuring instrument.

Table I shows soil properties measured in the two 40 ft. holes. While these soil properties are considered relatively accurate, some inaccuracy in moisture content was brought about by the impact method used by the driller to take the soil samples. The impact method of taking soil samples used by the driller produced a visible gradient in soil moisture. Evidently moisture was driven out of the samples with the impact method of taking soil samples. Since soil conductivity is heavily dependent upon soil moisture, we feel that the conductivity measurements are also inaccurate due to slightly lower moisture contents at the shallow depths and considerably lower moisture contents at the greater depths. Table 1 also shows conductivity of the soil samples assuming 0 moisture. With a few more temperature vs. depth measurements we should be able to use the wave propagation equation to calculate the soil thermal conductivity and diffusivity. One does this by varying the soil conductivity and diffusivity until the predicted wave temperatures are exactly equal to those being measured. This should give relatively accurate values for both conductivity and diffusivity.

These investigators are currently involved in another program directed toward determining the performance of a thermal envelope house. The house program is having a beneficial effect on the passive cooling program because many of the measurements and calculations are required for both programs. The thermal envelope house was monitored over the winter and has now been equipped with sufficient instrumentation to determine its performance in the cooling mode. Dry-Bulb temperatures are measured along the tube length and Wet-Bulb temperatures are being measured at the inlet and exit. Ground temperature probes are also located adjacent to the tube at intervals away from the tube. These measurements should permit us to verify the actual performance with performance predictions using the "Line Source" program.

Installation of the ground temperature probes at the thermal envelope house was accomplished with a fabricated water drill. The drill worked relatively well, although several minor problems were encountered. Since this drill is intended to be used for installation of some of the temperature probes in the passive field, we will correct the problems before the passive field instrumentation is installed.

We are presently scheduled to install the passive field the week of 28 July. We had originally intended to contract with a company to install the field but due to the small field size, most companies were reluctant to come out for the few hours necessary to install such a field. We also

TABLE NO. I SOIL PROPERTIES FROM TEST HOLES 1 AND 2

SAMPLE NUMBER	DEPTH Ft	DRY DENSITY Lbs/Ft^3	MOISTURE CONTENT % Dry Wt.	THERMAL CONDUCTIVITY BTU $Ft/Ft^2 hr.$ °F	HEAT CAPACITANCE BTU/$Ft.^3$ °F	THERMAL DIFFUSIVITY $Ft^2/hr.$
B.1.1	3.5 -5.0	83.1	12.1	.556	24.5	.023
B.1.2	8.5-10.0	61.4	27.1	.609	27.3	.022
B.1.3	13.5-15.0	80.8	22.1	.864	31.9	.027
B.1.4	18.5-20.0	86.1	25.2	1.052	36.6	.029
B.1.5	23.5-25.0	81.7	35.0	1.052	42.8	.025
B.1.6	28.5-30.0	79.2	35.0	1.015	41.5	.025
B.1.7	33.5-35.0	77.3	35.0	.970	40.5	.024
B.1.8	38.5-40.0	81.8	35.0	1.054	42.8	.025
B.2.1	3.5 -5.0	67.4	17.9	.590	23.8	.025
B.2.2	8.5-10.0	69.1	13.7	.476	21.5	.022
B.2.3	13.5-15.0	71.1	23.6	.727	29.1	.025
B.2.4	18.5-20.0	97.3	16.6	1.043	33.0	.032
B.2.5	23.5-25.0	84.2	35.0	1.167	44.1	.026
B.2.6	28.5-30.0	78.0	35.0	.969	40.8	.024
B.2.7	33.5-35.0	76.5	35.0	.937	40.0	.023
B.2.8	38.5-40.0	89.6	35.0	1.281	46.9	.027
Old Snow				.16	14	.01
Dry Sand				.1	19	.005
Wet Sand				1.0	25	.04

felt that it would be educational for our purpose to see what problems were encountered if we installed the field ourselves. We should be installing the field during the week of 28 July using graduate students and rented trenching equipment.

Equipment necessary for modeling actual loads on a house has been identified and ordered and should arrive the first week in August. It will then be connected to the field and the system put on line. The equipment selected for load simulation will permit us to exactly simulate the hour by hour load of the house under various operating scenarios. For our work we will be simulating the load of a well insulated low energy house located in Atlanta.

Work during the coming month will concentrate on getting the field in the ground, getting the load simulation equipment and instrumentation in place, continued simulation using the three simulation routines, continued monitoring of ground temperatures and verification of the simulation routines with the data measured at the thermal envelope house.

D-48-636

INVESTIGATION OF PASSIVE COOLING FOR HOT-HUMID CLIMATES

Progress Report
No. 6

Submitted to the

Research and Development Branch
Heating and Cooling, Conservation and Solar Applications
Department of Energy

College of Architecture
Georgia Institute of Technology
Atlanta, Georgia 30332

November 4, 1980

Project Director

JAMES M. AKRIDGE
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INVESTIGATION OF PASSIVE COOLING FOR HOT-HUMID CLIMATES

Work during this report period has concentrated on installation of the passive cooling field and the equipment and instrumentation necessary to simulate a load (building). Papers were also prepared for the Hybrid/Passive Conference in Washington and the ISES Passive Conference in Amherst.

The 700 ft. of 1½" diameter polyethylene pipe was installed at a depth of four feet. Original plans had been to lay the pipe in a 6" wide trench made by a powered trencher. This plan was abandoned after one day of effort when it was discovered the field site previously had been an old road and the curbstone had been left when the road was removed. The curbstone, at a depth of approximately 2 feet, proved to be an insurmountable obstacle to the trencher.

Since we had established that ground temperatures at a depth of four feet were too high to provide any cooling, those initial experiments with one uninsulated field appeared of little value. If the uninsulated experiments were not of value, use of the trenches was thus unnecessary. Use of the trenches was abandoned and a bull dozer was used to excavate the site to a depth of four feet. Once the site was excavated the field was laid out and the excavation backfilled to a depth of three feet. The site was then left for four weeks to permit the soil to settle and pack. Several good rains occurred during this time, assisting in obtaining a well-settled field.

After the four week settling period the site was hand-leveled with rakes, covered with 2" of Dow SM extruded polystyrene rigid insulation. The insulation was covered with a 6 mil polyethylene film. All joints in the film were taped with polyester tape to minimize moisture migration through the field.

The insulation and film were then covered with one foot of dirt which had previously been removed from the hole. This was packed and then planted with a mixture of rye and fescue grass. Every effort is being made to establish a very good grass cover for the field to reduce solar heating of the soil above the field.

The field was located so that one of the previously installed 40' deep soil temperature wells was located in the center of the field midway between the serpentine field coils. The field was heavily instrumented with premium grade copper constantan thermocouples. Figure 1 shows the sensor location within the field. Figure 2 is a section through the field showing the relationship between the insulation, surface, and coil.

The building load simulation equipment has been purchased and installed. Figure 3 shows a schematic of the equipment which will be used to program and simulate the building load. Figure 4 is a flow schematic showing how the flow directions change, depending upon whether the field is being charged or discharged.

Work throughout the program has been well-documented with 35 mm slides. Sixteen slides which give a good summary of this documentation are being shipped under separate cover. The sixteen slides are numbered and correspond to slide copies which appear as figures S1 -- S16 in this report.

Figure S1 shows the site as it appeared before the start of construction. This slide shows the north side of the architecture building and the drilling rig which sank the two 40 ft. wells now used to monitor soil temperature vs. depth.

Figure S2 shows a temperature probe constructed of PVC pipe and copper-constantan thermocouples being inserted into the well which is located within the test field. The probe is shown in a "broken" state. As each section was inserted, the next section was aligned and solvent bonded. This eventually resulted in a straight probe that is forty feet long. A second probe was sunk in the shade of the elm tree shown in the background. The probe is located about where the center of the truck is located in the figure.

Figure S3 shows the field laid out in chalk before the start of excavation. One should notice that at this time the ground had very little grass cover and is in direct sunlight.

Figure S4 shows the top end of the forty foot probe flush with the ground surface. The thermocouple bundle, including the ground surface thermocouple, can be seen exiting from the tube. The interior of the tube was filled with fine sand and the hole around the tube was filled with a 50-50 mixture of bentonite and portland cement.

Figure S5 shows the site during excavation. The size of the excavation (4'deep x 40'wide x 60'long) can be estimated by the size of the bulldozer, driver, and car.

Figure S6 shows the field laid out in chalk on the bottom of the excavation. Since the far end is slightly lower than the near end, a crosswise serpentine was used rather than a lengthwise configuration. This configuration was chosen to reduce air entrainment and pumping losses.

Figure S7 shows a closeup of a pipe fluid temperature thermocouple. The couple is taped to the exterior of the pipe and then covered with flexible urethane pipe insulation to insure that the temperature measured more closely approximates that of the fluid rather than that of the adjacent ground.

Figure S8 shows the pipe superimposed on the chalk layout. The figure illustrates the closeness with which the pipe was made to conform to the desired layout. Adjacent pipe spacing is four feet.

Figure S9 is a closeup of the pipe bends showing that the 1½"d polyethylene tubing bends quite readily to the 2 ft. radius without buckling.

Figure S10 shows the bulldozer covering the pipe. By covering the pipe from one side, it was possible to cover an area and then drive over the area with the bull dozer to cover other areas.

Figure S11 is a closeup of the bundle of thermocouples coming from one of the forty foot ground temperature wells.

Figure S12 shows a water drill being used to sink a shallow vertical temperature well. This drill has been used to sink wells to depths of 20 ft. Approximately 15-30 minutes were required to drill to the 20 ft. depth.

Figure S13 shows the leveled field just prior to the installation of the rigid styrefoam insulation. The water drill can be seen at the far end of the field.

Figure S14 shows the insulation partially in place and the first steps in the final covering. The student is shown sealing the area where the vertical temperature probe extends through the insulation.

Figure S15 shows the 6 mil polyethylene being stretched over the rigid insulation. UV protected polyethylene was not used because the one foot of soil covering it filters out all UV.

Figure S16 shows the field being covered with the last foot of soil. If the insulation was covered to a foot depth beginning on one side, the small rubber-tired dozer could be driven over the covered area to cover other areas without damaging the insulation.

The air-to-water heat exchanger has been installed and final hookup of the instrumentation is being made. Work during the coming month will concentrate on checkout of the system using the air-to-water heat exchanger in a natural convection mode. Subsequent tests will be directed toward determining the most cost-effective forced convection approach. Work will also be directed toward continued computer simulations of the load, charge and discharge system and initial studies of the building side heat exchanger.

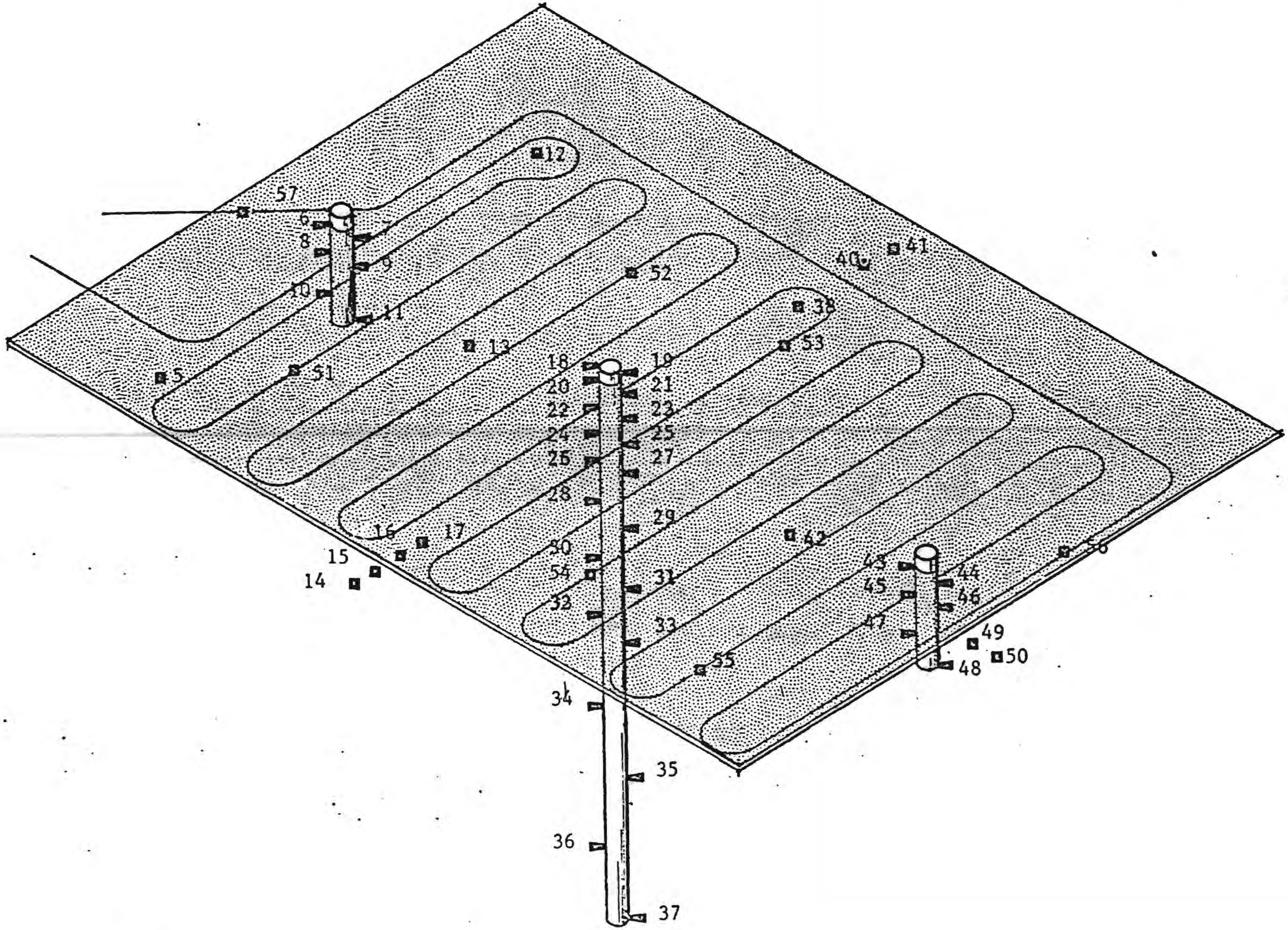


FIGURE 1: SENSOR LOCATION SCHEMATIC

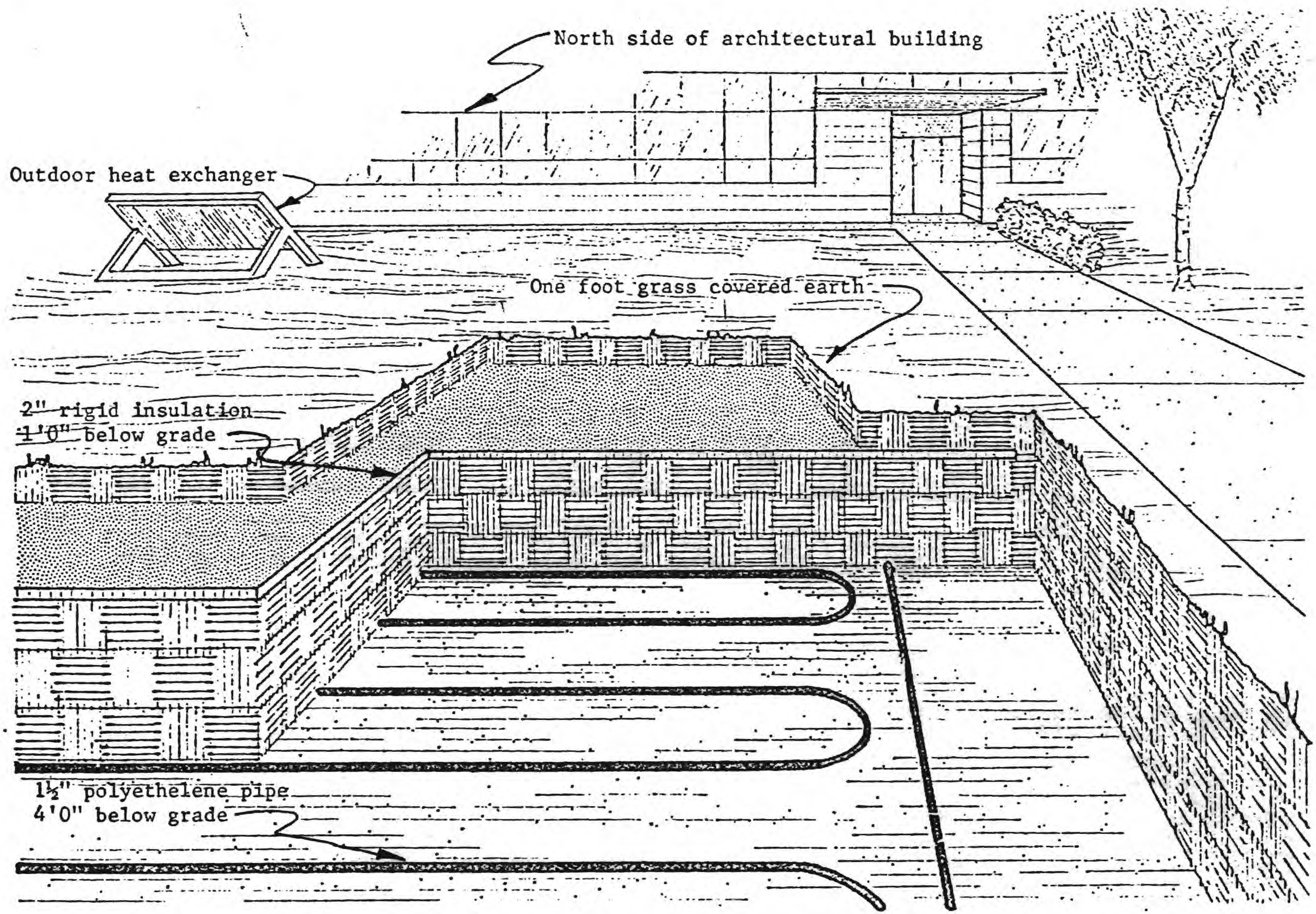


FIGURE 2 : COOLING FIELD SECTION

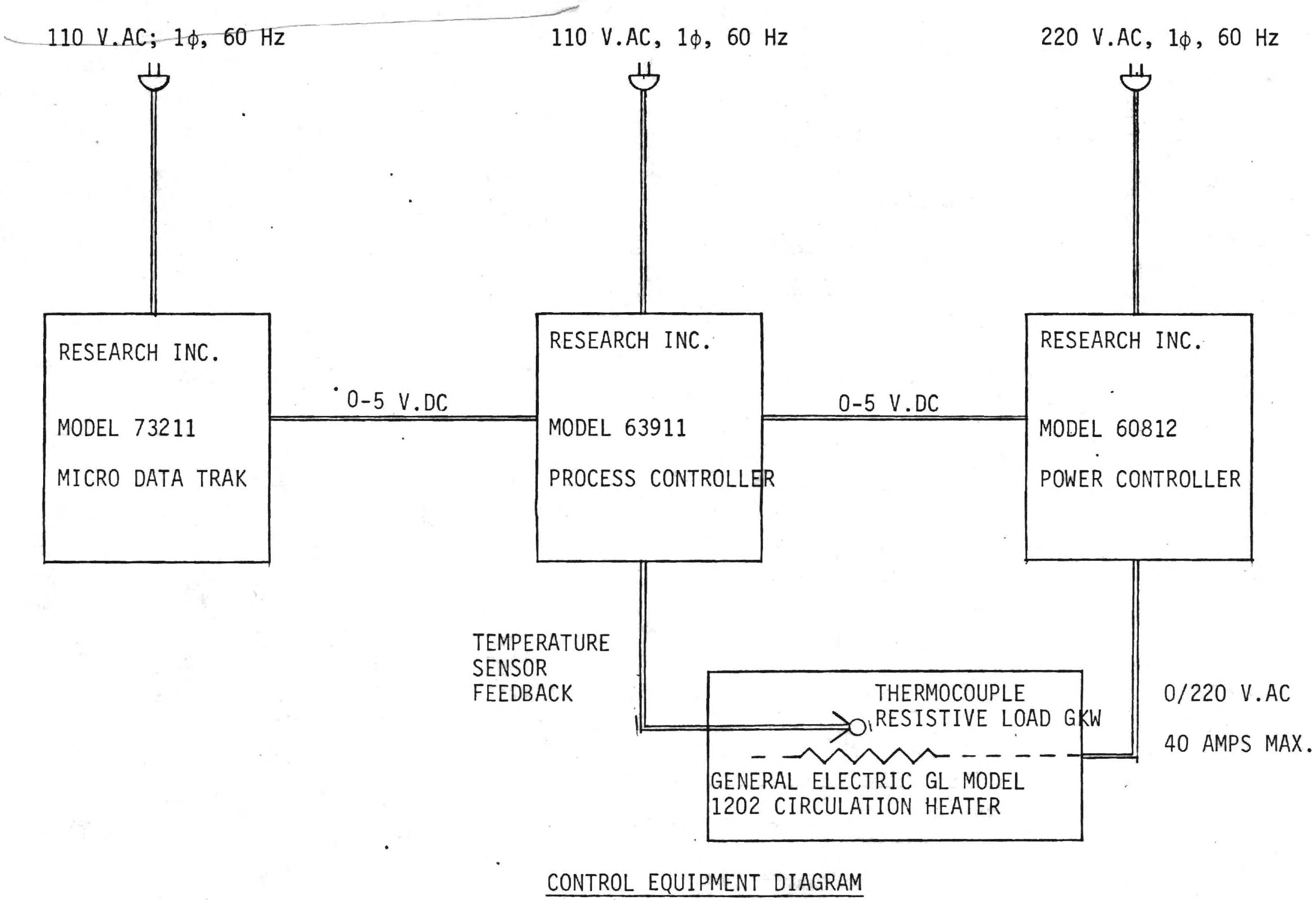
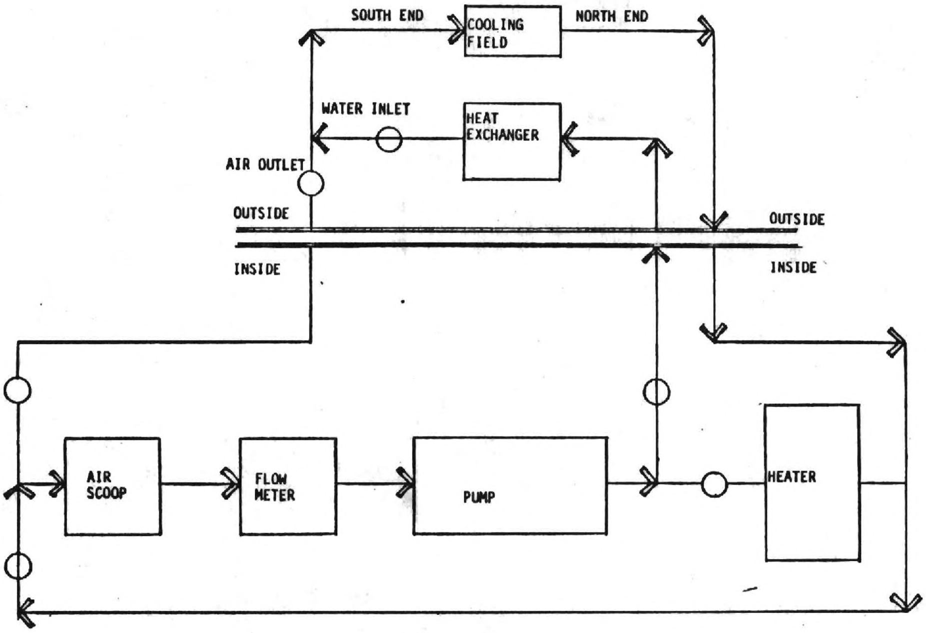


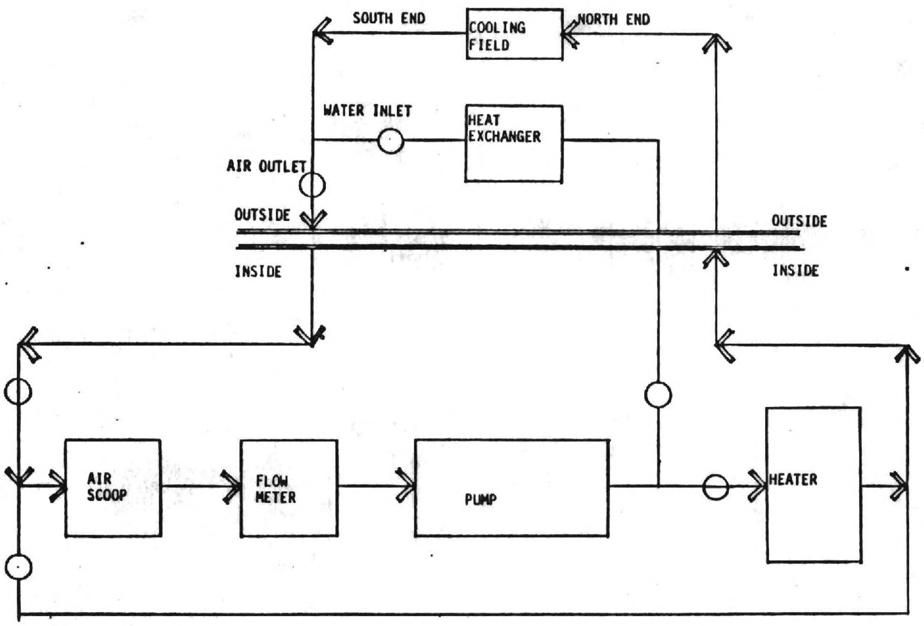
FIGURE 3 Building Load Simulation Equipment Schematic



PIPING DIAGRAM

COOLING FIELD AND PUMPING STATION

WINTER MODE - CHARGING



PIPING DIAGRAM

COOLING FIELD AND PUMPING STATION

SUMMER MODE - COOLING

FIGURE 4: COOLING FIELD AND PUMPING STATION - FLOW DIAGRAMS

Figure 51. Drilling 40 ft. Vertical Temperature Well.

Figure S2. Installation of Vertical Temperature Probe.

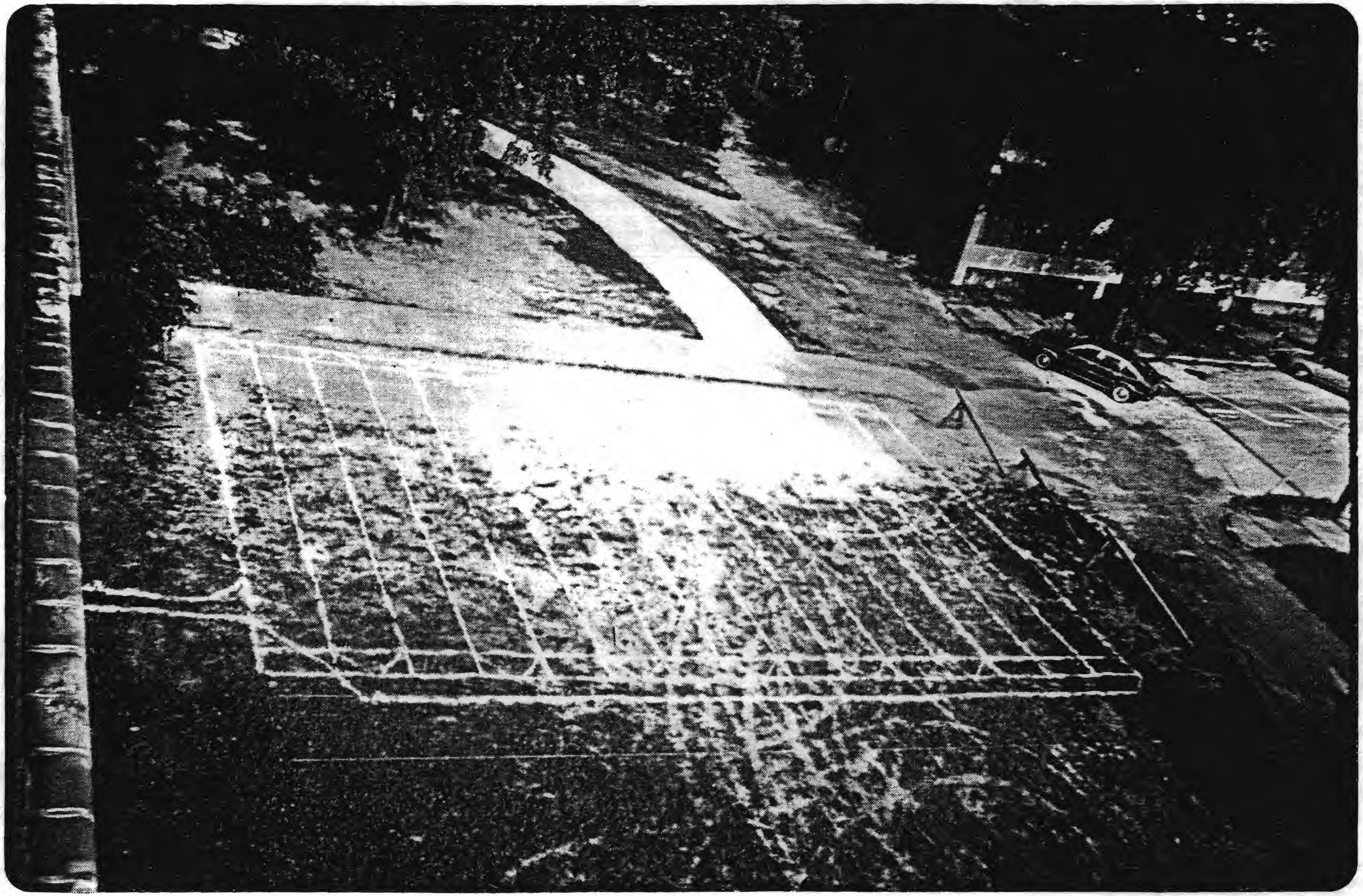


Figure S3. Layout of Field before Excavation.

Figure S4. Vertical Probe Upper End Showing Thermocouples.

Figure S5. Field Excavation.

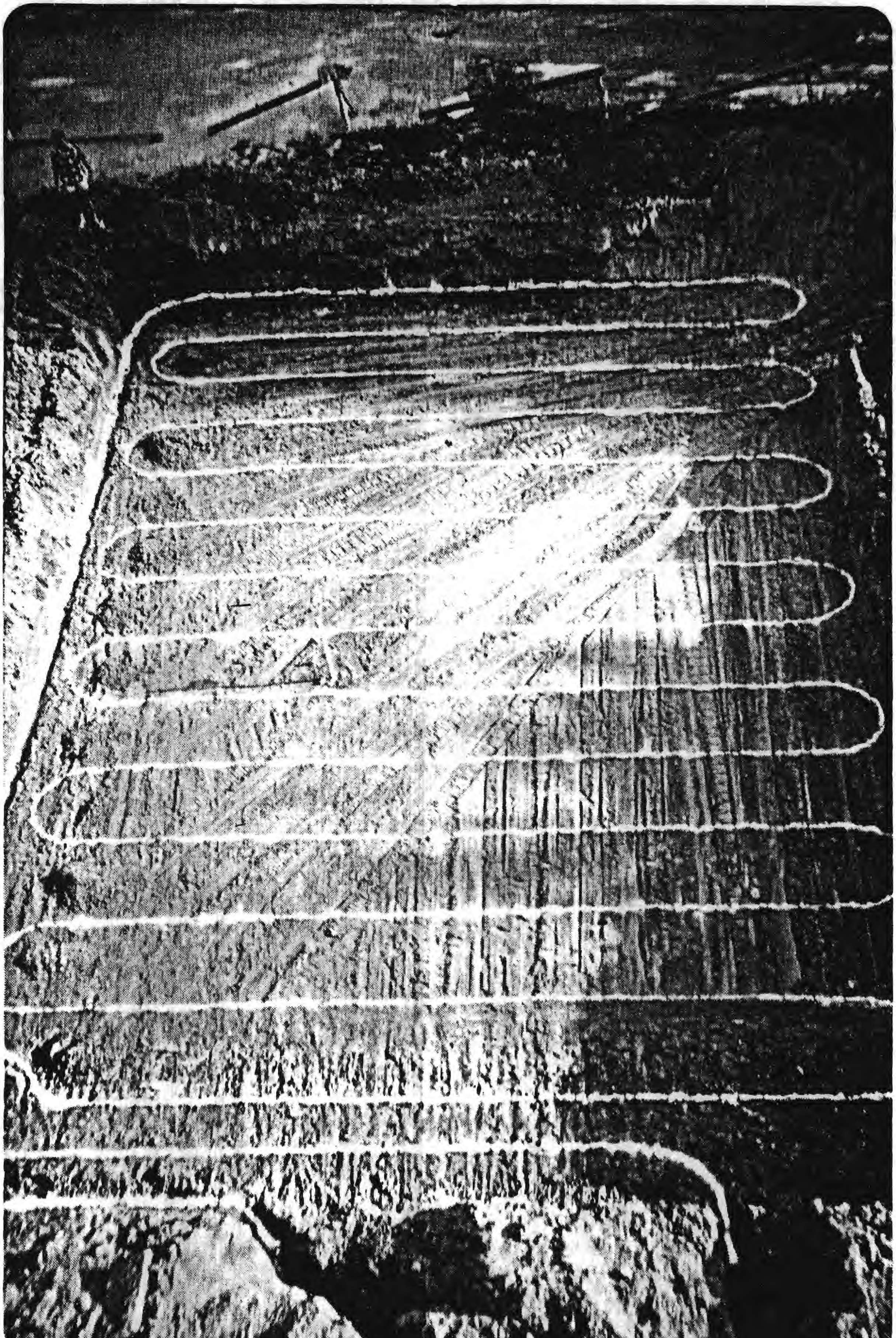


Figure S6. Chalk Layout of Field on Excavated Site.

Figure S7. Polyethylene Tube Showing Fluid Temperature Thermocouple.

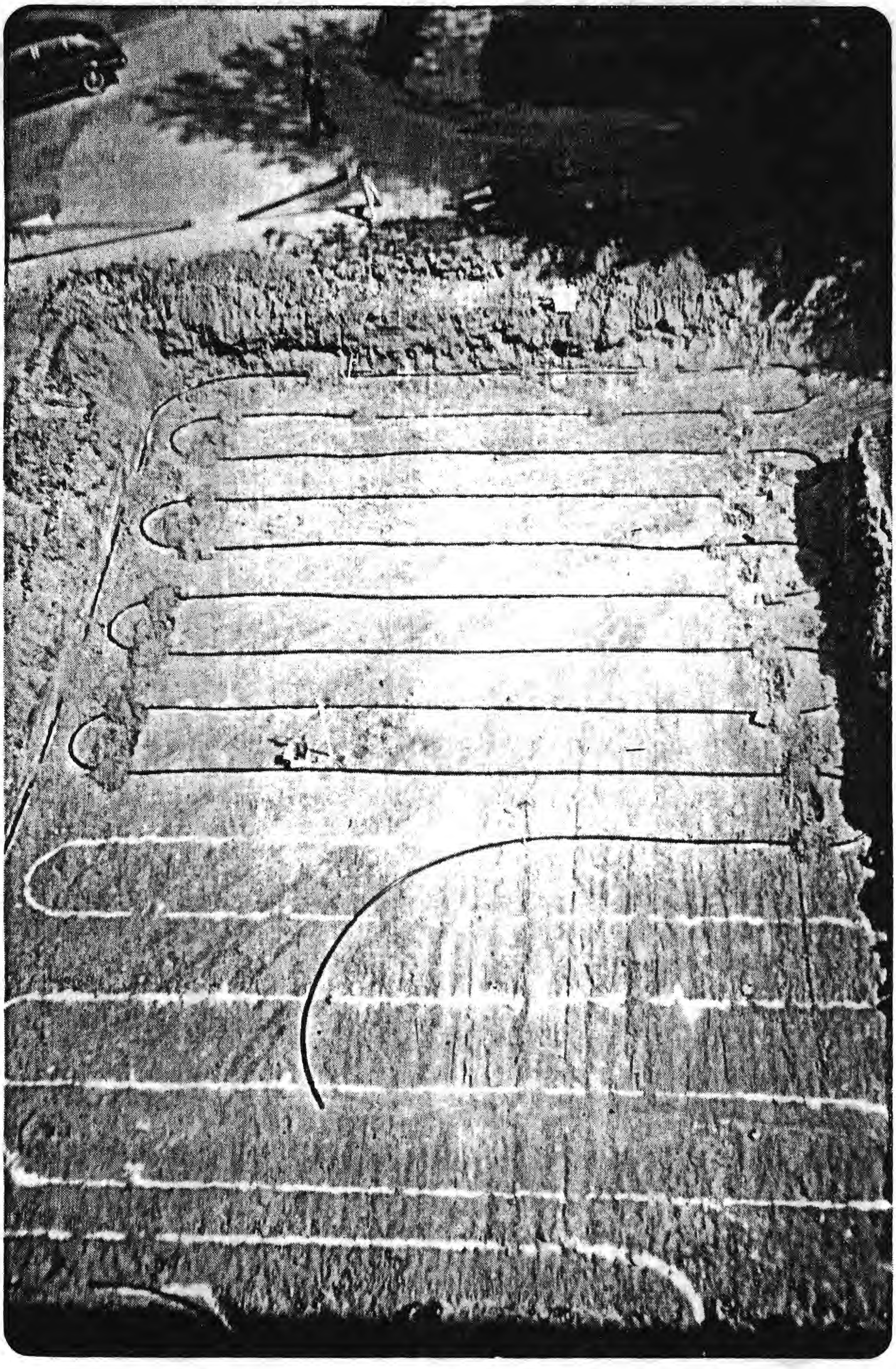


Figure 58. Field Layout Showing Tube Conformity to Desired Configuration.

Figure S9. Closeup of Field Bends Showing Lack of Buckling.

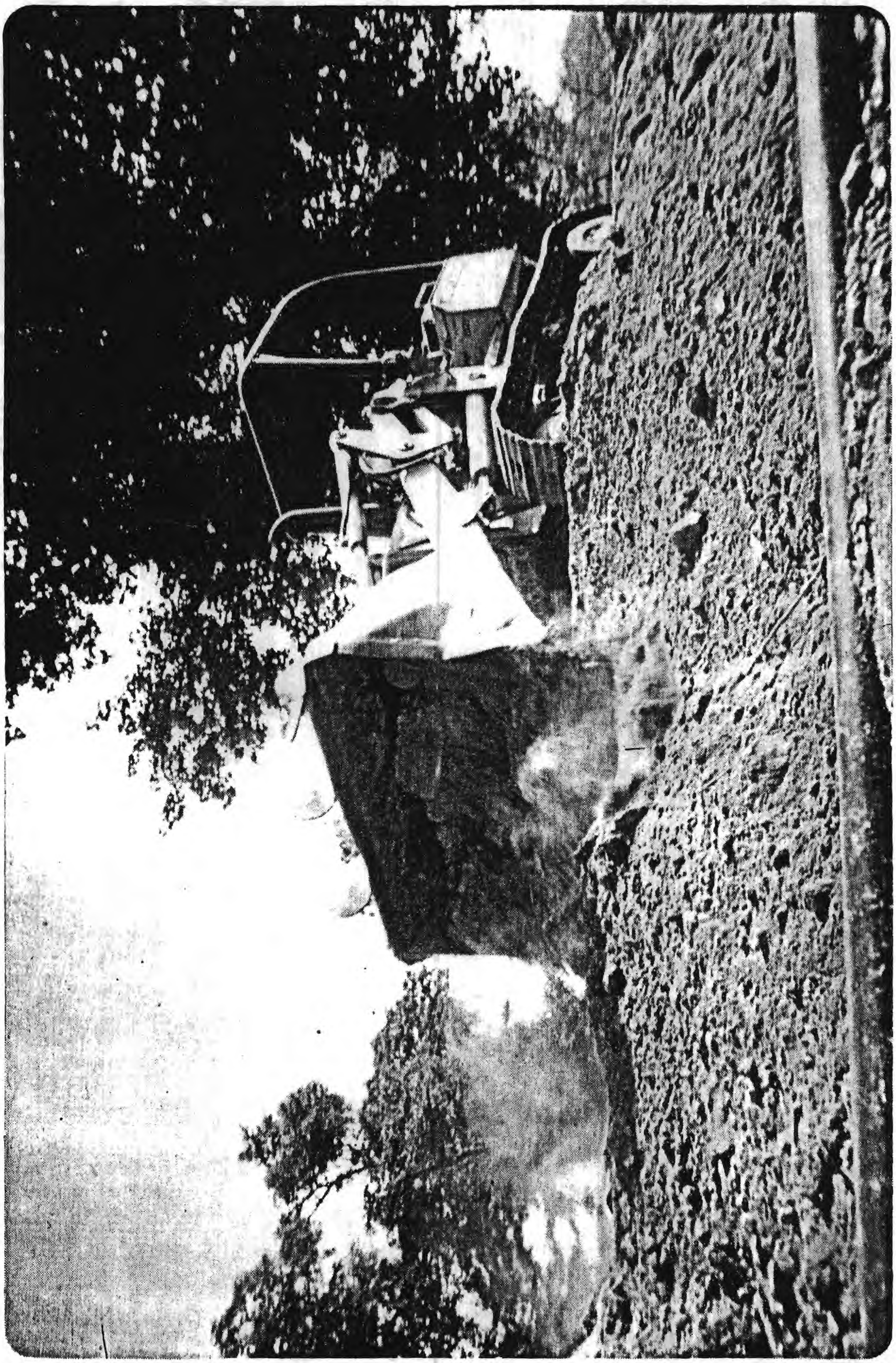


Figure S10. Closeup of Coil Being Covered.

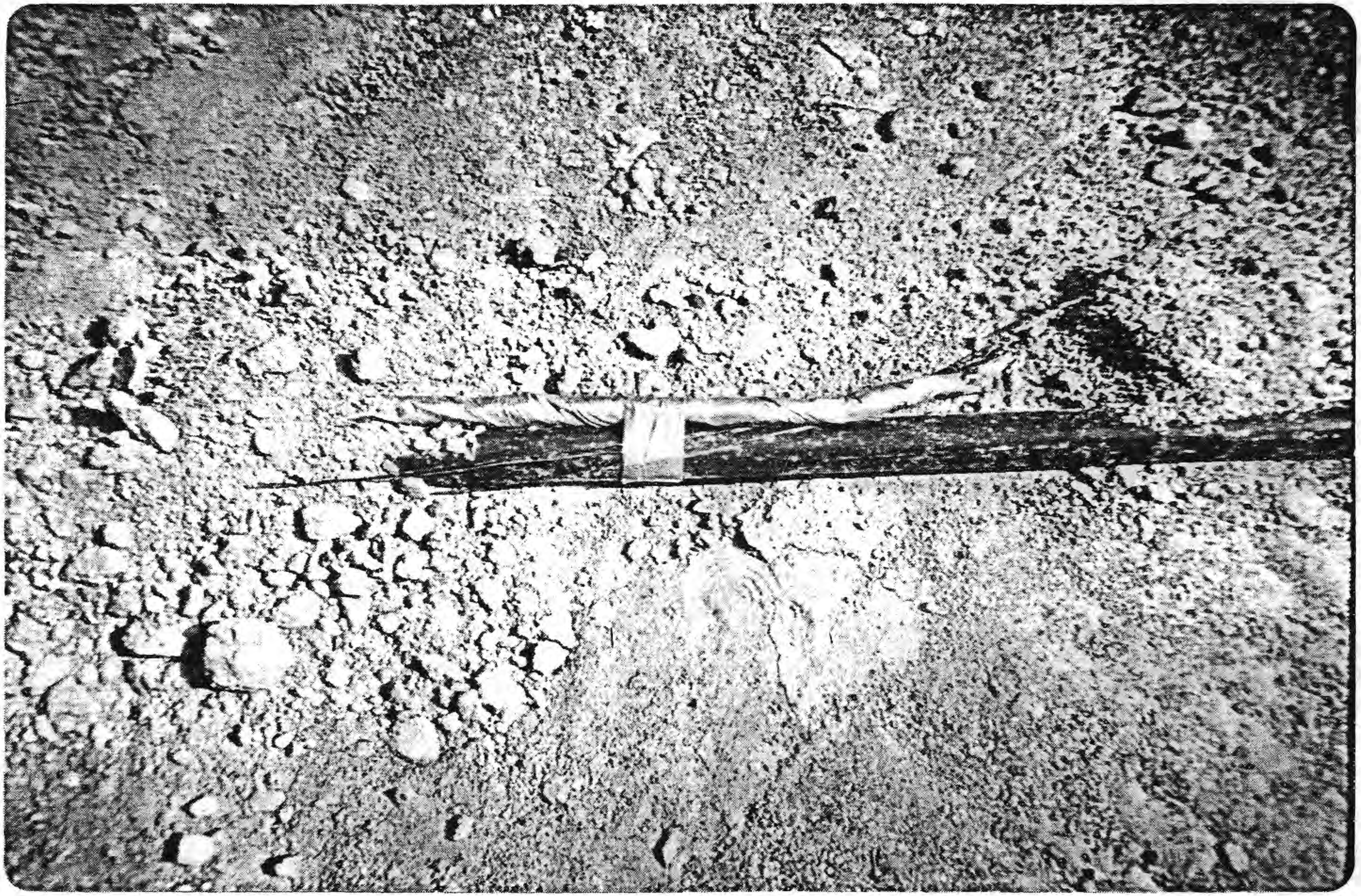


Figure S11. Closeup of Vertical Probe Thermocouples.

Figure S12. Water Drill Being Used to Drill Vertical Probe Well.

Figure S13. Covered Field Prior to Installation of Insulation.

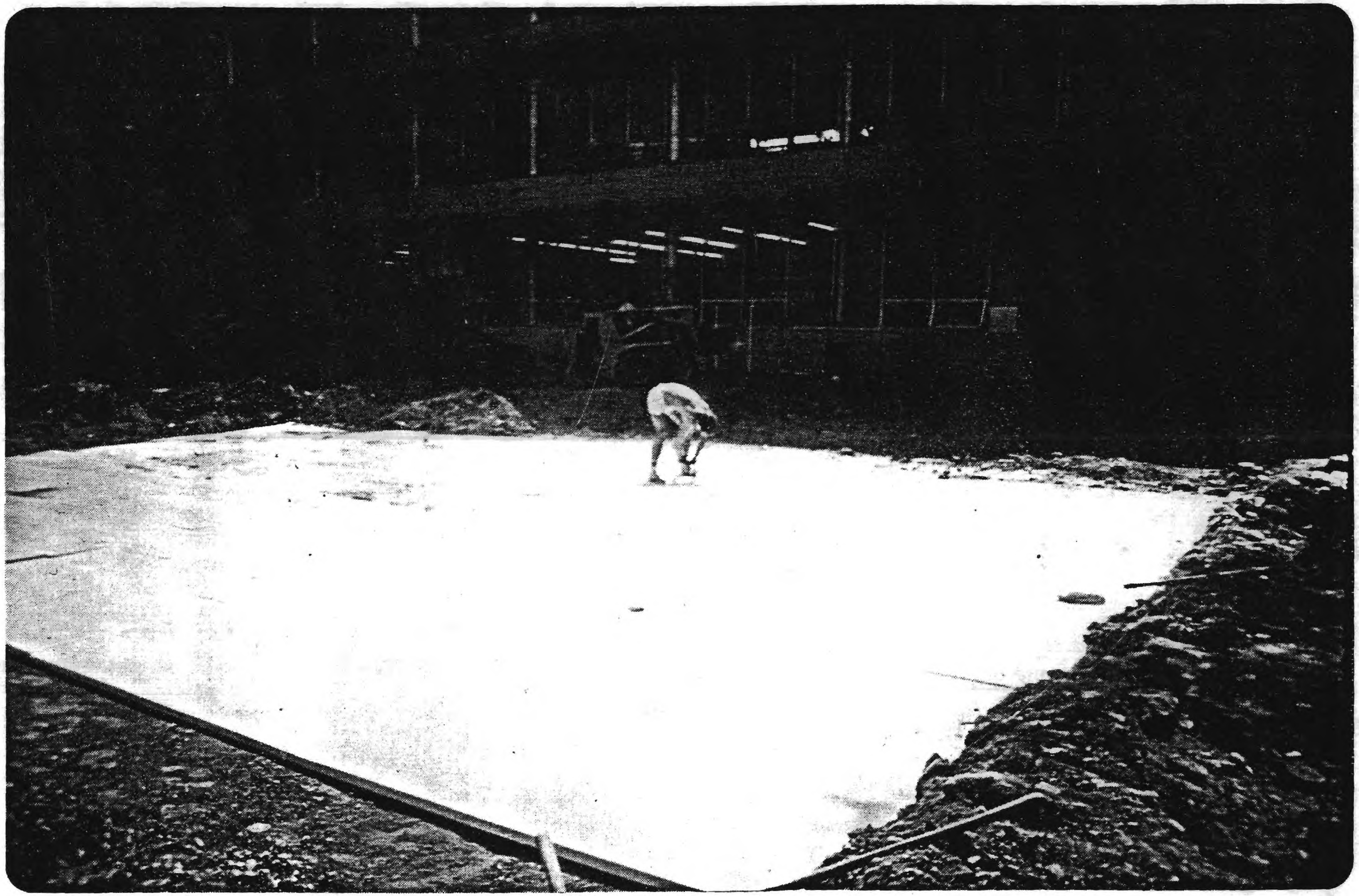


Figure S14. Field Showing Insulation Partially Installed.

Figure S15. Field Showing 6 mil Vapor Barrier.

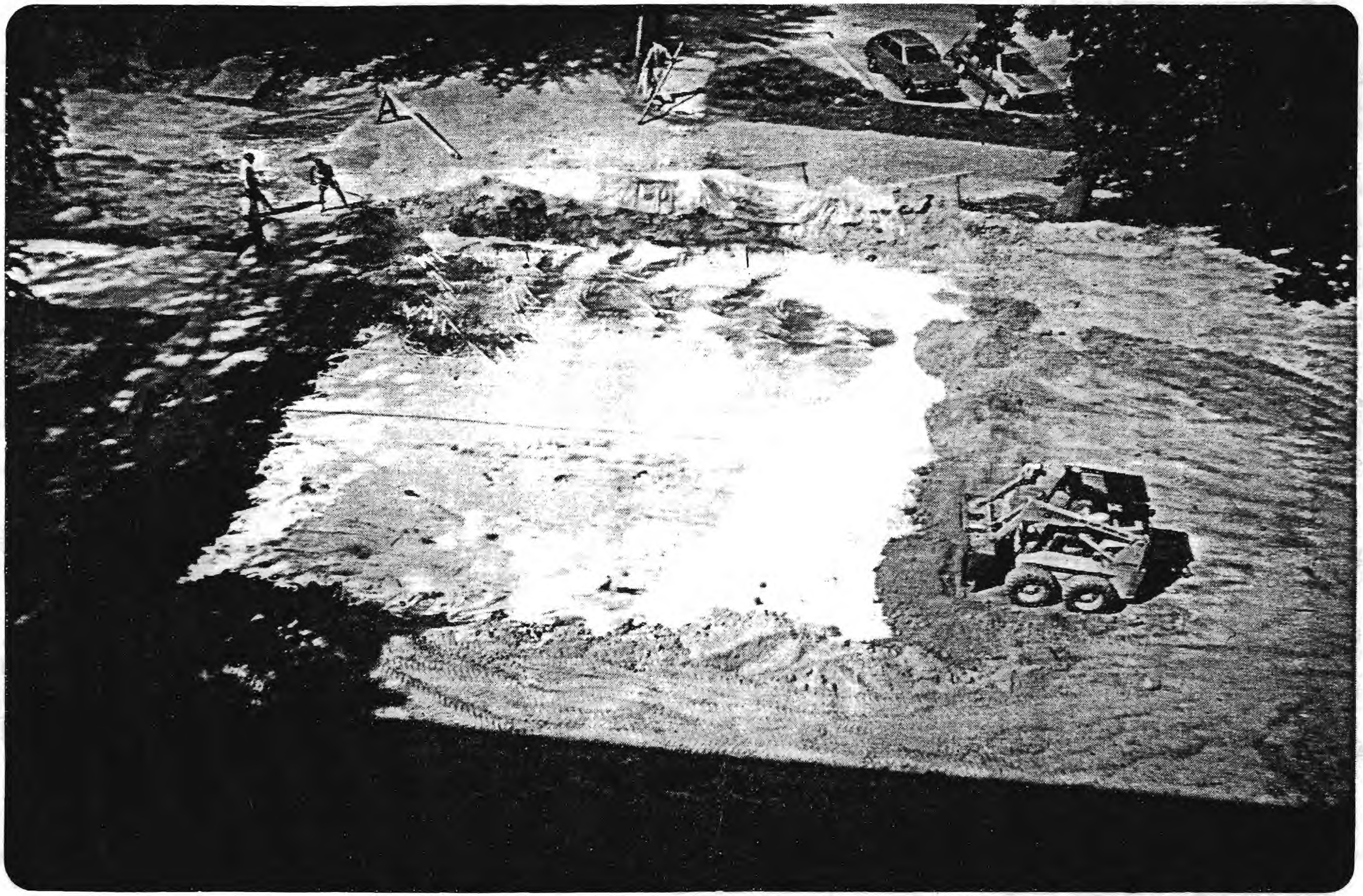


Figure S16. Field During Final Stages of Covering.

D-48-636

INVESTIGATION OF PASSIVE COOLING FOR HOT-HUMID CLIMATES

Progress Report
No. 7

Submitted to the

Research and Development Branch
Heating and Cooling, Conservation and Solar Applications
Department of Energy

College of Architecture
Georgia Institute of Technology
Atlanta, Georgia 30332

6 March 1981

Project Director

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INVESTIGATION OF PASSIVE COOLING TECHNIQUES FOR HOT-HUMID CLIMATES

Work during this report period has concentrated on checkout of the cooling field and instrumentation and the investigation of the field charging rate using both natural and forced convection on the air to liquid heat exchanger. Building side heat exchanger strategies are currently being investigated through computer simulation using the MITAS multinode computer program and on inhouse twenty-five node program using the GROCS film flow equivalent of the individual tube flow. The panel and room cooling test facilities are being designed.

Minor problems were encountered in getting the above ground piping sealed. Minor leaks kept causing loss of water and subsequent air locking of the water pump. These have all been solved and the system is now performing as designed.

Significant problems were encountered with the instrumentation used to read temperatures throughout the field. This problem was very much a surprise because all of the probes measured properly when measured individually with a high accuracy portable thermocouple readout. When all of the thermocouples were connected to the data acquisition system as recommended by the manufacturer, 10 to 15 thermocouples gave readings which were slightly in error while 3 to 4 gave readings which were 10-15^{o}F in error. This problem was finally traced to ground loops passing through a floating ground in the data acquisiton system. Since many of the thermocouples are buried in ground with a finite thermal conductivity, the ground loop problem surfaced when more than one thermocouple was connected to the floating ground. The problem was solved by isolating each of the thermocouples from the floating ground with a zener diode. Fluke has recommended an improved (from the standpoint of lightning protection) modification using varistors from each thermocouple to a hard ground. The improved protection will be implemented before we enter the thunderstorm season.

Since we felt it undesirable to pump antifreeze solutions through the underground piping until we were positive we had control of all leaks, the system has been operated with water throughout this winter. We had established that until the field temperature began to approach freezing, the water was unlikely to freeze if the pump kept the water circulating. It was also obvious that the water would freeze if the pump were to quit when the outside temperature was below freezing. The most probable cause of water not flowing was identified as pump stoppage caused by power interruptions. The pump and differential temperature control are therefore driven by a power inverter running off a battery. The battery is kept charged with a line-powered battery charger. This system keeps the pump running at all times when the ambient temperature is below freezing.

FIELD CHARGING

Since the success of seasonal storage of cooling potential is highly dependent upon keeping the energy expended in charging the storage to a minimum, the field was initially charged using only natural convection over the heat exchanger. This proved satisfactory until the field temperatures dropped below 55°F at which point the charging rate became unacceptably low. Two 185-watt 2200 CFM fans were added to the heat exchanger in late

January to increase the charging rate. This modification increased the charging rate to acceptable levels although the fan capacity vs. power consumed vs. energy stored has not been optimized. This optimization will be accomplished before the next charging cycle.

Progress report number six pointed out that the field had been installed during the hottest part of the summer. The field was also insulated toward the end of the hot weather. This was done to meet the time schedule for winter charging and not because it was the optimum time (it is the worst time one could install the field). By excavating to four feet, we permitted the earth at a depth of four feet to be heated to surface temperatures. We were measuring surface temperatures of 135°F at this time. When the field was covered and insulated we trapped a block of earth that was at an average temperature considerably higher (10 to 15°F) than nondisturbed earth at that depth. The problem was compounded by not solving the charging circuit problem until late November. This meant that the earth we wanted cool was very hot and was not being cooled either by the field or the surface wave as the ambient temperature began to drop during the early part of the winter. This has resulted in the field has been at a temperature higher than soil at a similar depth that is not in the field since the beginning of the active charging. The difference between the soil temperature at 4' within the field and soil at 4' not within the field has been steadily decreasing throughout the charging cycle. They have not yet become equal.

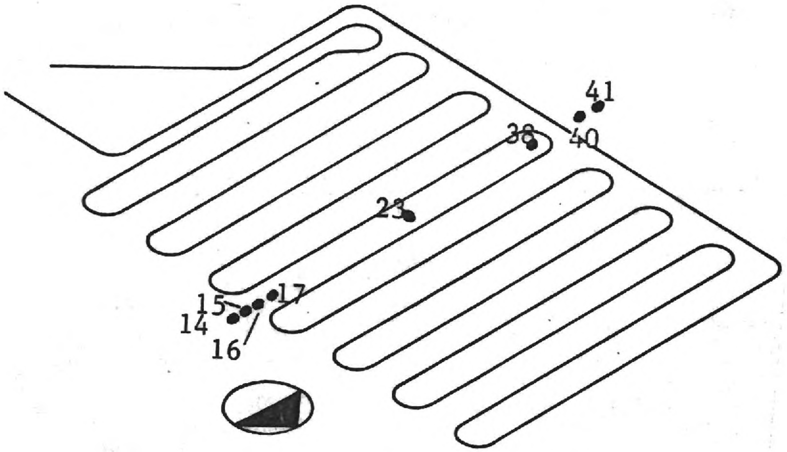
Figure 1 shows the temperature distribution across the field at a four foot depth for three different dates. One will notice the significant drop in field temperature between 27 January and 18 February and the decrease in temperature difference between the field and non-field is evident along either edge of the field. The increase in field temperature between 18

February and 24 February resulted from a very warm spell when the ambient temperature was above the field temperature and the charging system was not operating. The temperature increase resulted from migration of energy from the soil above and below four feet during this period.

Figure 2 shows the temperature distribution along the field center at the four foot depth for the same dates shown in Figure 1. The importance of the data shown in Figure 2 is the lack of significant temperature gradient from the field inlet end to the exit end.

Figure 3 shows the temperature distribution from the surface to a depth of forty feet for both the test hole (that located in the cooling field) and the reference hole located about 56 feet to the west. Notice the significantly lower temperatures in the reference hole for both dates than for the test hole. This difference results primarily from the reference hole being located in the shade of a tree while the test hole was in an open field with no ground cover. This illustrates the importance of ground cover for ground used for passive cooling purposes. The lower temperatures at the surface and one foot depth for the test hole in February results from the insulation at the one foot depth decreasing energy flow from the lower depths.

Work during the next report period will concentrate on optimizing the building side heat exchanger and evaluation of options available. Design of the panel and room test facilities will be completed and construction started on the panel test facility. The field will continue to be charged when ambient conditions permit.



Sensors are located in the plane of the pipe field 4'-0" below finish grade.

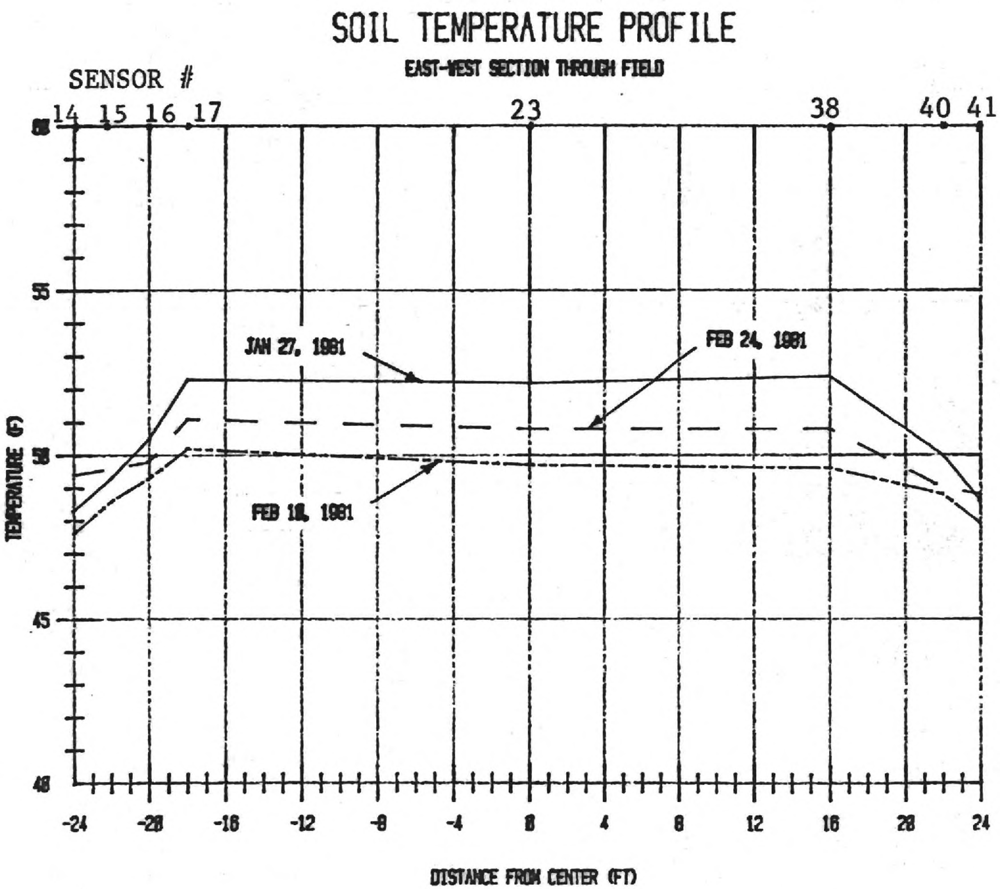
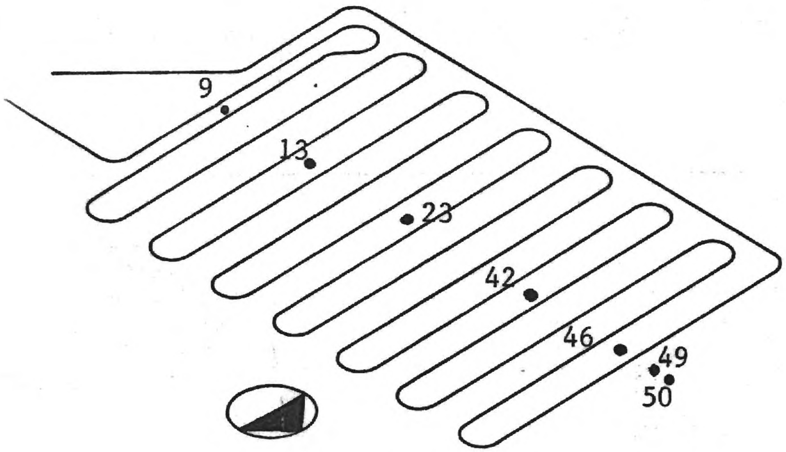


FIgure 1. Temperature Distribution Across the Field.



Sensors are located in the plane of the pipe field 4'-0" below finish grade.

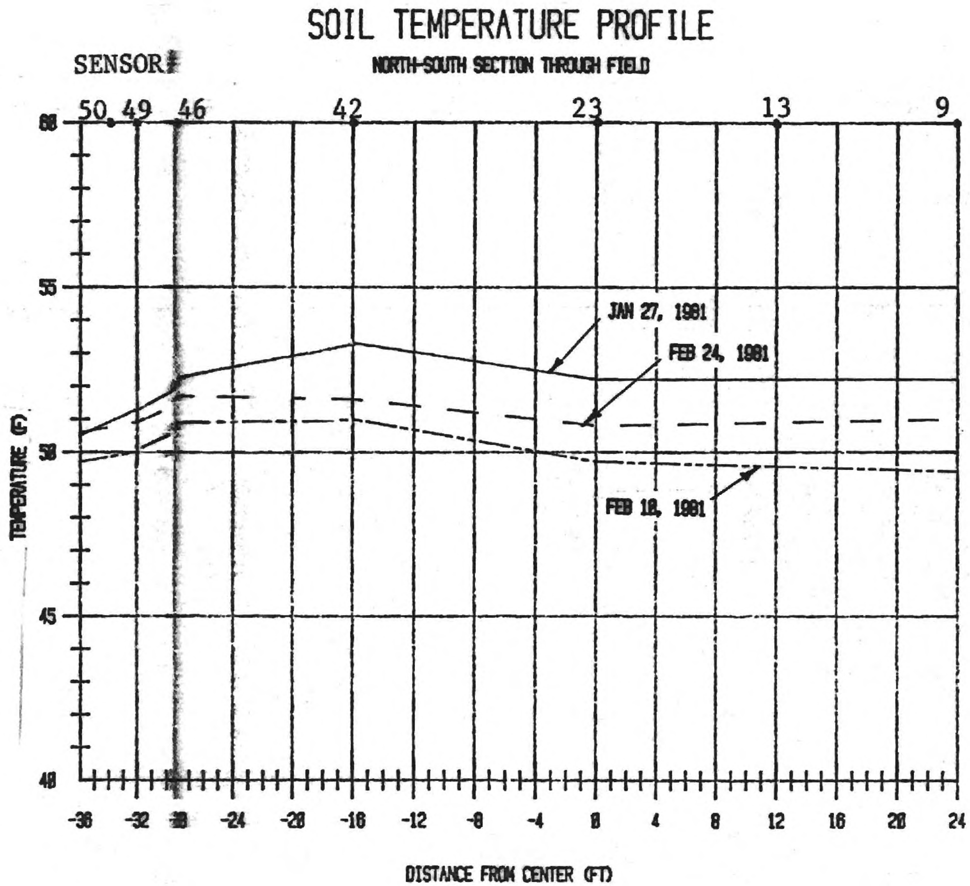


Figure 2. Temperature Distribution Along the Field.

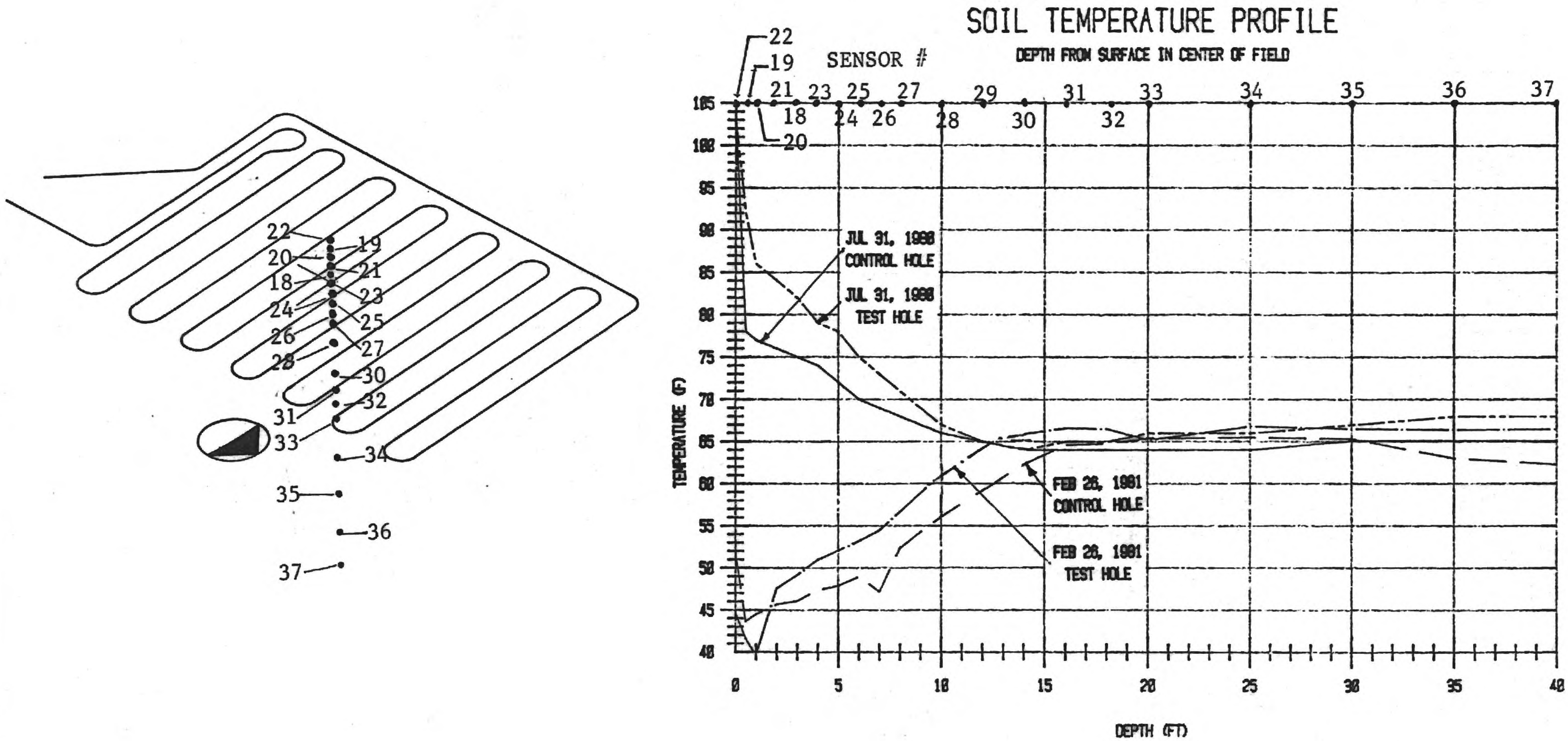


Figure 3. Temperature Distribution vs. Depth for Test and Reference Holes.

D-48-636

INVESTIGATION OF PASSIVE COOLING FOR HOT-HUMID CLIMATES

Progress Report
No. 8

Submitted to the

Research and Development Branch
Heating and Cooling, Conservation and Solar Applications
Department of Energy

College of Architecture
Georgia Institute of Technology
Atlanta, Georgia 30332

25 May 1981

Project Director

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INVESTIGATION OF PASSIVE COOLING TECHNIQUES FOR HOT-HUMID CLIMATES

Work during this report period has concentrated on investigating and optimizing the building side heat exchanger and on construction of the panel test facility. The above-ground plumbing for the coil field has also been modified extensively to improve performance, increase reliability and to simplify construction.

The field differential thermostat has demonstrated its value during the intermediate weather when ambient temperatures are well above the field temperature during the day but drop below the field temperature for several hours during many nights. When the field is above ambient, the differential thermostat turns on the pump and starts cooling the field.

Although the field temperature beneath the insulation has risen significantly from the low experienced in early March, it is now substantially below the temperature at a four foot depth in the open area adjacent to the field and is also below the temperature at the four foot depth in the control hole. Figures 1, 2 and 3 show how the vertical temperature distribution has changed since the field installation was started last summer.

Since the ground water temperature is approximately 62°F, the field is expected to warm quite rapidly until it approaches the 62°F range due to energy diffusing into the field from greater depths. At first glance this was disturbing due to the rapid rise being seen at the four foot depth. When one realizes that the larger block of earth below the four foot depth is actually being cooled below the temperature it would normally have, the temperature rise becomes less of a concern.

When the field layout and above-ground plumbing was first designed, it was thought that there might be a significant advantage in passing the water through the field in one direction while charging the field and in the reverse direction when operating in the cooling mode. This turned out to be insignificant due to the relatively long time available for the energy to diffuse through the field. This layout also made it very difficult to operate the air-to-liquid heat exchanger in series with the field and the building side heat exchanger. Simultaneous operation of all three exchangers is desirable when the wet-bulb temperature is lower than the field temperature and when cooling is desired at the building side exchanger.

The above-ground plumbing was extensively changed during this reporting period to permit series operation of the three heat exchanger (ground coil, building side, and air-to-liquid). Figure 4 shows the new plumbing layout. Comparison with Figure 4 in Progress Report No. 6 shows the new layout to be much simpler in addition to having much less pumping losses.

When it is desirable to operate the air-to-water heat exchanger in series with the other two heat exchangers, the three-way valve closes the by-pass and passes the water through all three exchangers. If evaporative cooling boost is desired, a second solenoid valve opens a line connecting city water to spray nozzles located on the air-to-liquid heat exchanger. This allows the system to take advantage of relatively low wet-bulb temperatures in early summer.

Optimization of the building side heat exchanger through computer simulations has proven to be much more difficult than expected. A 100 node model representing the area from one pipe to the line halfway between pipes was constructed for the MITAS[1] simulation. A similar 32 node model was also designed for a T-NODE[2] program that will run on micro computers (Radio Shack, Model II in this instance). The simulations were in close agreement. Node number was held constant and the node size increased as the tube spacing increased. Again, the results from both the programs agreed quite well with both programs predicting that the temperature on the wall air side directly opposite the tube decreased as the tube spacing increased. Intuitively, this appeared incorrect so the T-NODE program was re-run using an increased number of nodes as the spacing increased. This showed the air side temperatures to vary as expected.

Careful examination of the modeling technique immediately around the tube was found to be inexact although not greatly in error. Both MITAS and T-NODE runs are being prepared using a more exacting method of modeling the area immediately around the tubs. The results from these runs will be reported on in the next progress report.

The panel test chamber has been designed and is nearing completion. Figures 5 and 6 show two views of the test chamber. The chamber has been designed with 6" thick foamed polyurethane (R-54) walls to minimize the effect of edge or wall heat losses. Due to the great weight of the test panels, the panels are suspended from a gantry by 1/8" steel cables shown in Figure 5.

Figure 7 shows the first concrete test panel scheduled for test in the next several weeks. The panel was built with a very large number of temperature sensors and the versatility to permit a variation of tube spacing through the opening and closing of several valves. The panel was also designed to be changed in thickness through the addition of one or more additional pours of concrete.

The full size test room has been designed as two separate test rooms with an interchangeable separating wall. This particular configuration allows one to change the separating wall by swinging one room away while the wall is changed and then swing it back into place while the tests are conducted. This configuration lends itself well to the study of convection and radiation coefficients as well as the radiative cooling potential.

Owing to the difficulty with instrumentation (reported in Progress Report No. 7) and the problem with the computer modeling as well as perhaps an overly optimistic schedule, portions of the program are presently about two months behind schedule. Fortunately, due to the separate nature of the tests planned over the next several months, part of the program will be on schedule while others catch up.

Work during the next report period will be directed toward solving the computer modeling problem, completion of the panel test box, construction of the full size test room and operation of the cooling field in the cooling mode.

The programmed operation of the cooling field in the cooling mode is scheduled to begin on June 1 using measured cooling loads from one of Georgia Power's "Good Cents" houses.

Work is also being directed toward determination of the most efficient auxilliary cooling device and mode of operation. Initial calculations using a water-to-water heat pump with an elevated evaporator temperature (65^{o}F) shows a very high coefficient of performances. High evaporator temperatures become possible when the heat pump is used as a source of chilled water for a radiative cooling panel.

JMA:SME

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1. "Martin Marietta Interactive Thermal Analysis System," Martin Marietta Corporation, Denver, Colorado, 1974.

2. Wright, Scott, T-Node "Thermal Network Simulation program". Thesis for Masters of Architecture, Georgia Institute of Technology, June 1981.

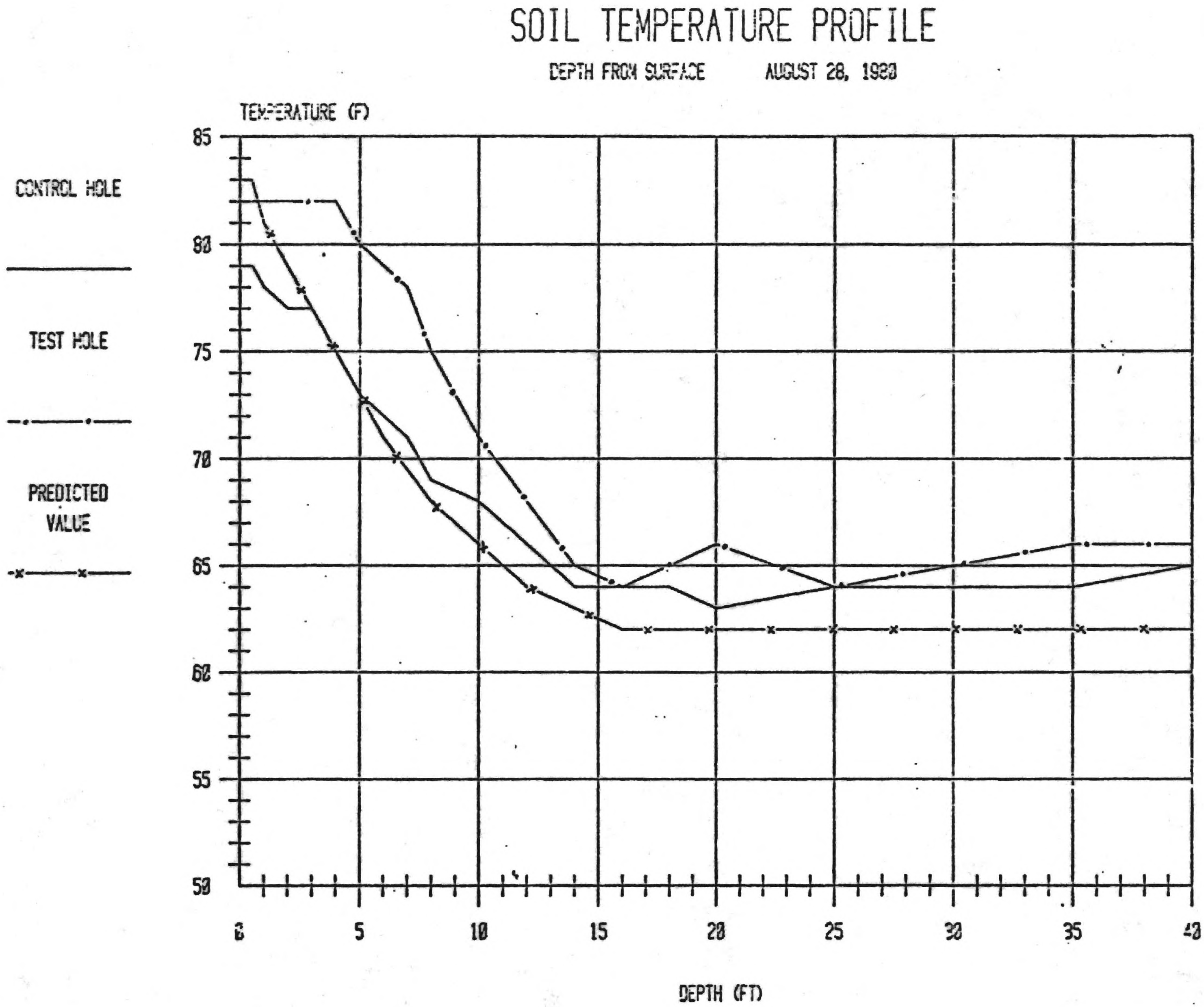

FIGURE 1. Soil Vertical Temperature Distribution Before Field Installation

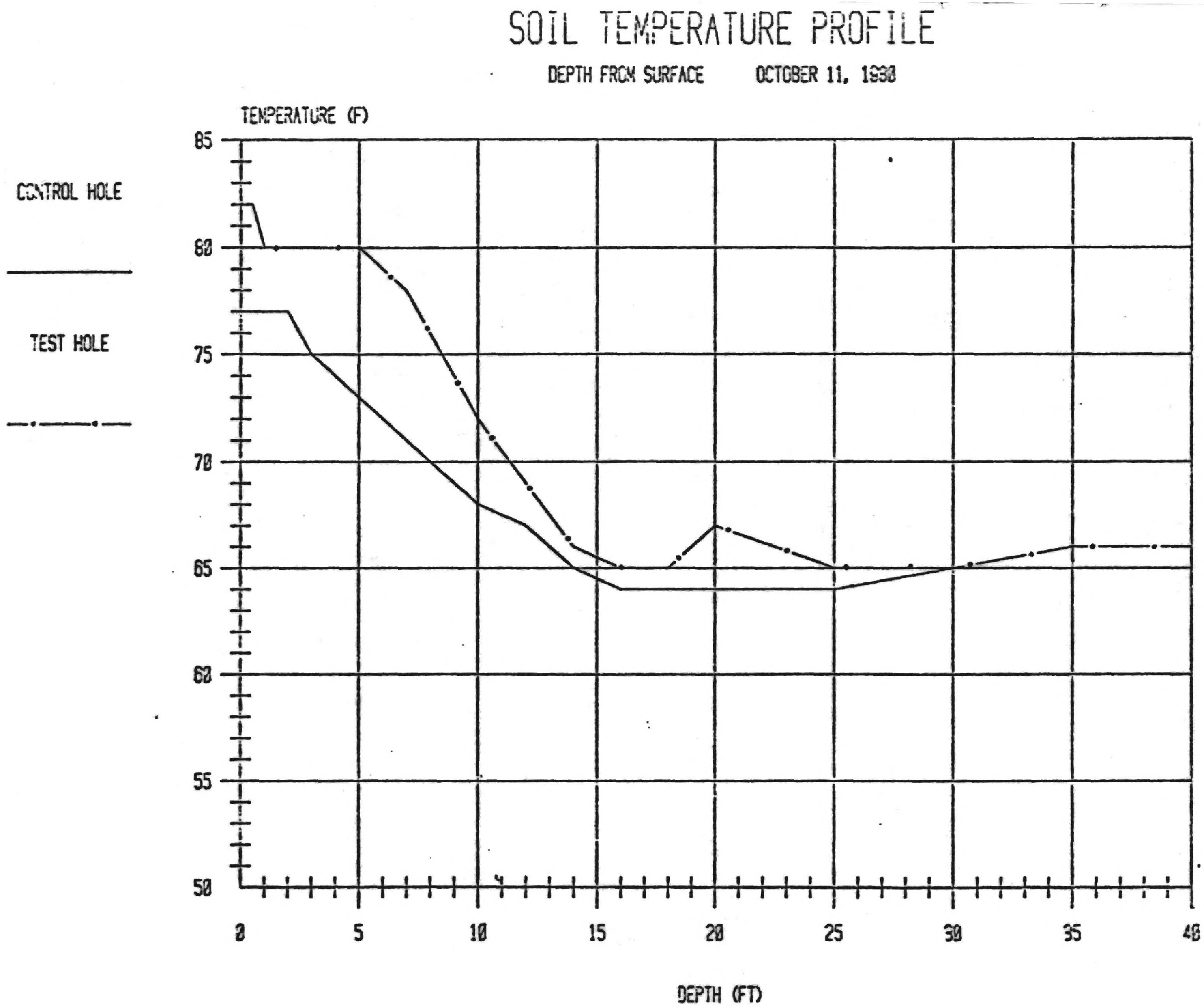

Figure 2. Soil Vertical Temperature Distribution after Field Installation

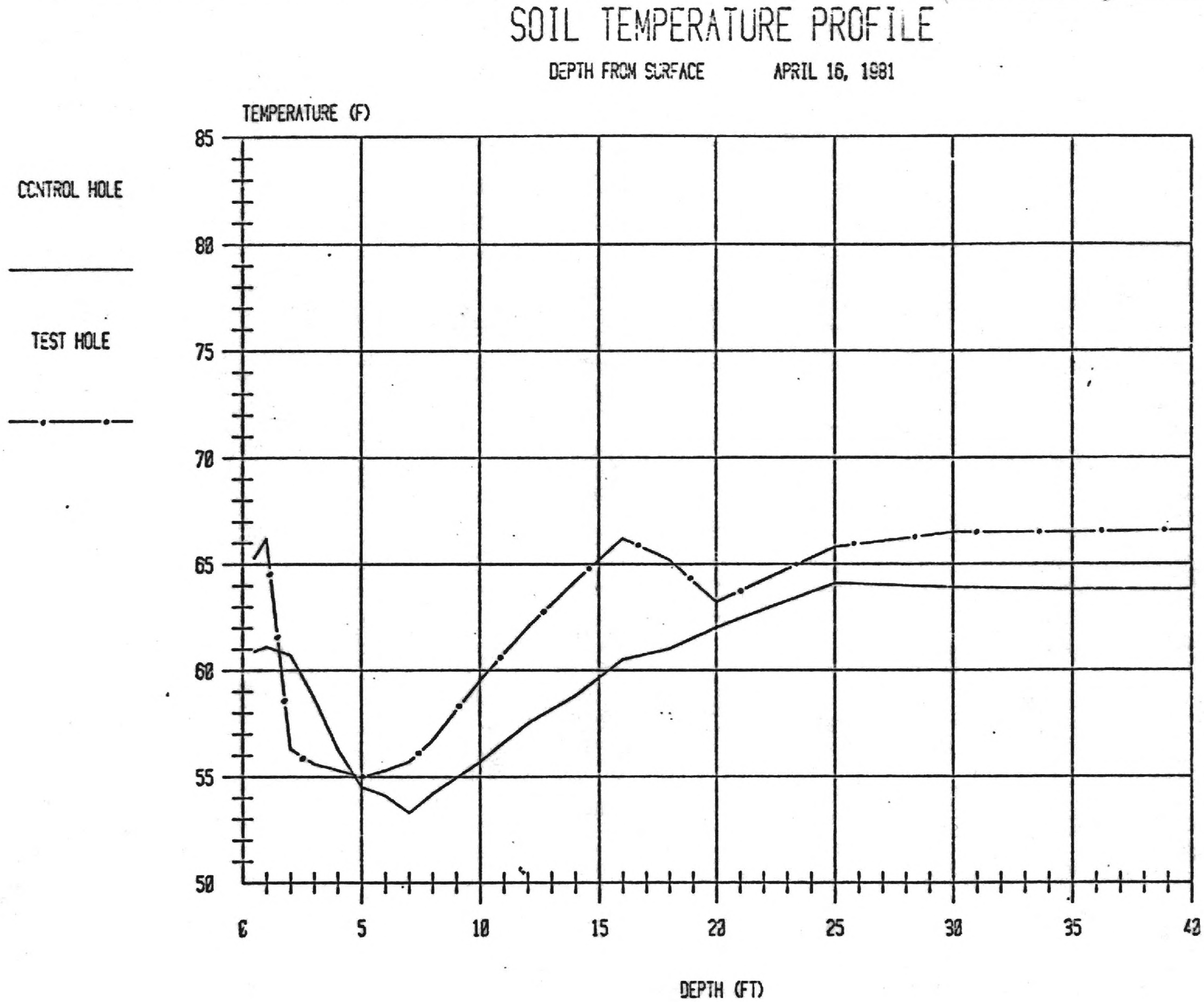

Figure 3. Soil Vertical Temperature Distribution at the End of Winter

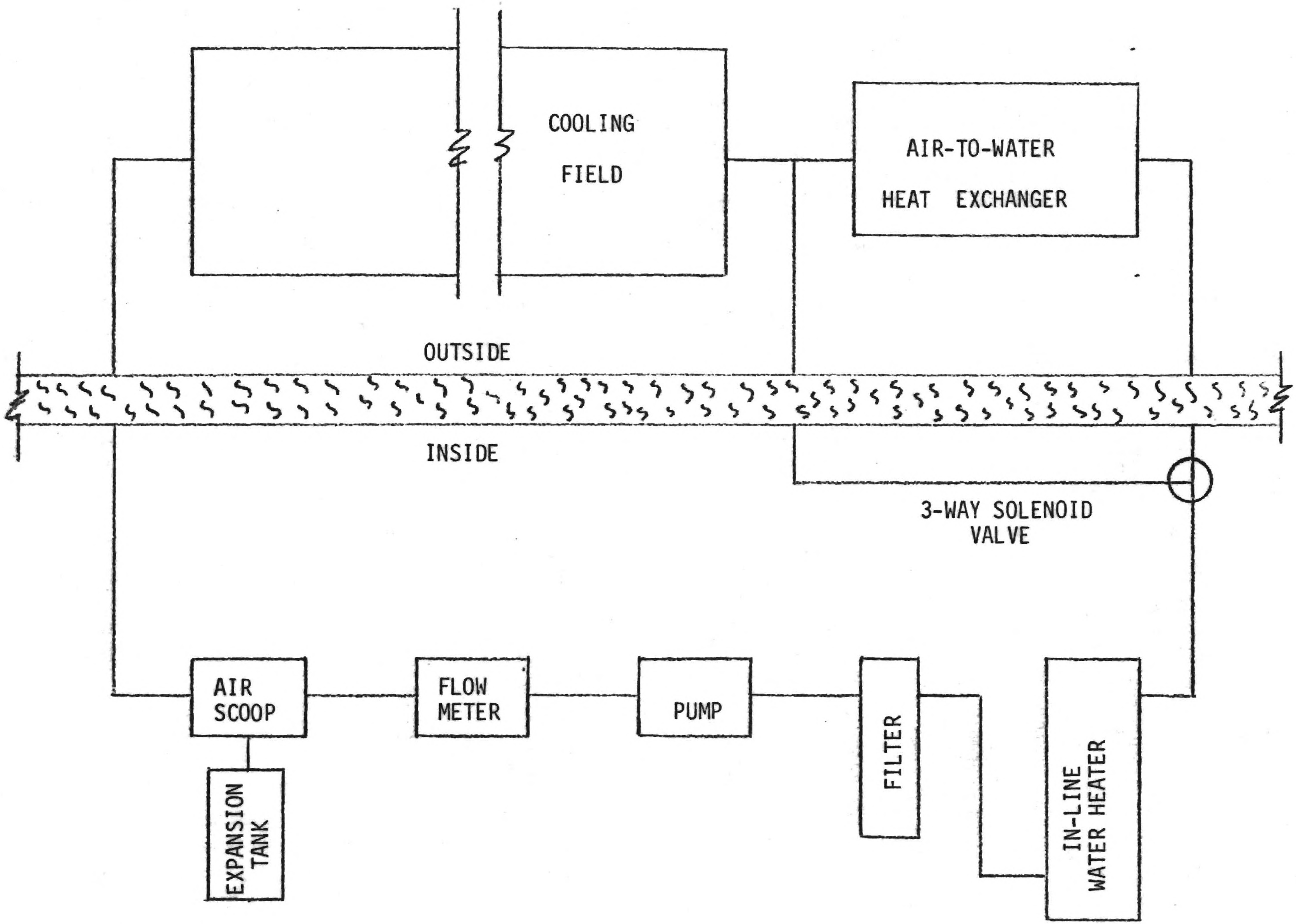

Figure 4. Cooling Field and Building Simulation Schematic

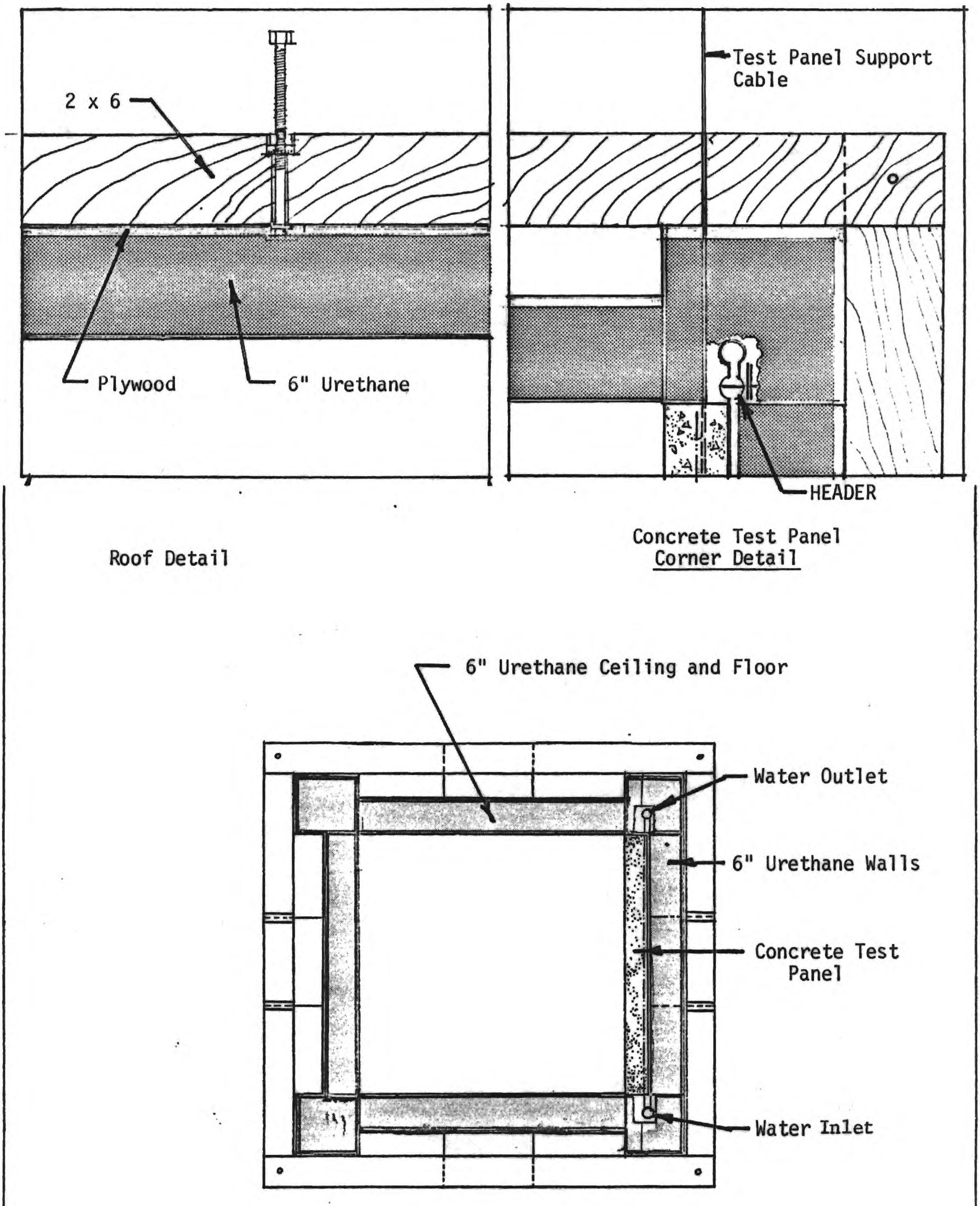

Figure 5. Section Through Panel Test Box

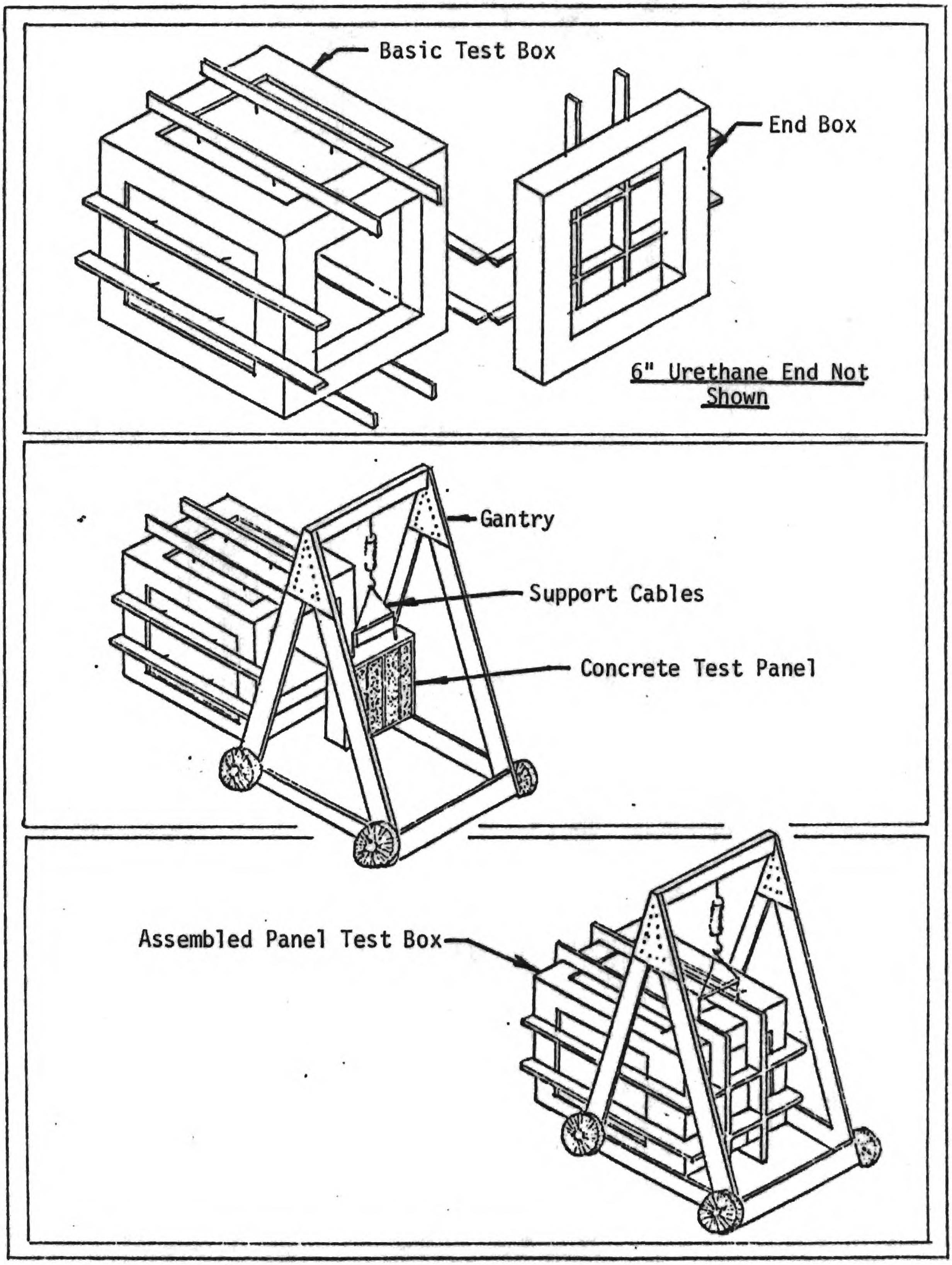

Figure 6. Panel Test Box - Exploded and Assembled Views

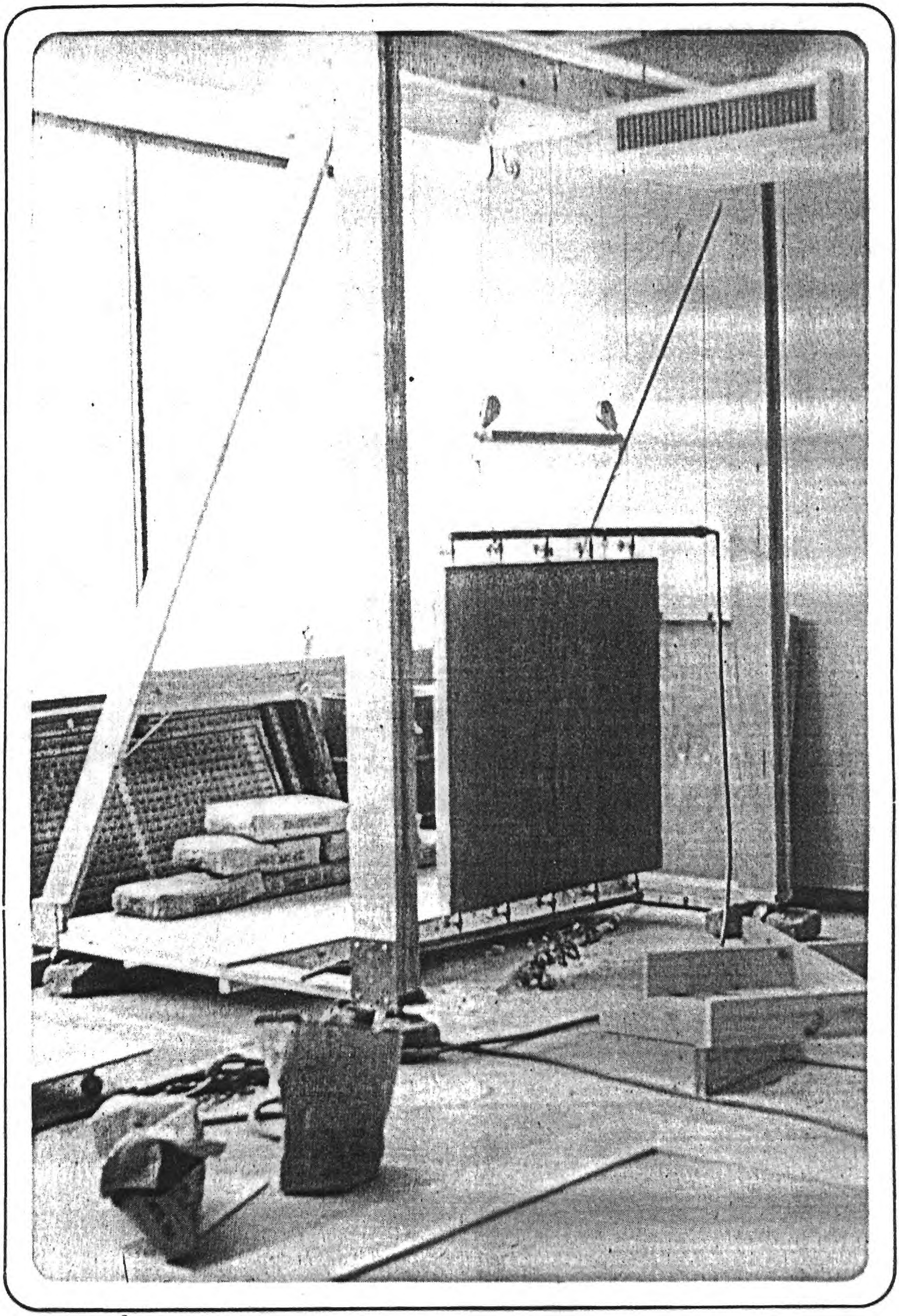

Figure 7.   Concrete Radiative Cooling Panel and Gantry

D-48-636

INVESTIGATION OF PASSIVE COOLING FOR HOT-HUMID CLIMATES

Progress Report
No. 9

Submitted to the

Research and Development Branch
Heating, and Cooling, Conservation and Solar Applications
Department of Energy

by the

College of Architecture
Georgia Institute of Technology
Atlanta, Georgia 30332

1 September 1981

Project Director

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INVESTIGATION OF PASSIVE COOLING TECHNIQUES FOR HOT-HUMID CLIMATES

SUMMARY

Work during this report period has concentrated on operation of the cooling field as a heat sink to carry a cooling load programmed into our fielding load simulator and to evaluate the performance of both lightweight and heavyweight cooling panels with several different tube spacings. Considerable effort has also been directed toward definition of the complete passive/active cooling system with emphasis on arriving at systems which compliment each other. Preparation for and attendance at D.O.E. program reviews has also been heavy during this report period.

Actual measured electrical consumption for a Georgia Power "Good Cents" house located in Columbus, Georgia was converted to hourly sensible cooling loads for each of the cooling months using the seasonal EER given by the manufacturer of the home's air conditioner. These energy profiles are given in figures 1 through 5 in both BTU/hour and kilowatts. Only sensible loads are used because the radiative cooling method employed in the detached earth tempering concept is not capable of carrying the latent load. Sensible load was estimated by dividing the total hourly load by 1.3.

The daily load profile for a given month was programmed into a Research Incorporated Model 73211 Micro Data Trak load programmer which controlled a Research Incorporated Model 63911 process controller with a 40 amp solid state switch. The solid state switch varied the power going to a General Electric 220 volt, 6 kw electric circulation water heater according to the programmed load.

Water was circulated through the buried cooling field and then through the water heater before returning to the field. The field was considered to be capable of passively cooling a building until the water temperature coming from the field rose above 72°F. The choice of 72°F was arbitrarily chosen because performance data for the radiative cooling panels were not available when those tests were conducted.

Table I shows the monthly sensible cooling load of the house, the cooling load carried by the field and the percentage of the sensible load carried by the field. While the percentage carried passively at first might seem low, one must go back to the start of the field charging to get a true picture of the cooling potential of this concept. Due to program scheduling it was necessary to install the field during the record heat wave being experienced in Atlanta in August of 1980. This resulted in the ground at the four foot depth being exposed to much higher temperatures than the four foot depth would normally experience. To minimize settling the field was left exposed for over four weeks after it had been back filled to one foot below grade. This permitted all of the earth within the field and around the field to reach much higher temperatures than normal. Again, due to scheduling the field was insulated with 2" of Dow extruded polystyrene in late September when the field and adjacent ground was very hot. The insulation was covered with a final foot of earth backfill.

The remainder of the charging circuit, the load simulation and the control strategy could not be installed until the field was in place, insulated and the instrumentation completed. The field was not ready to start charging until the first week in January at which time the ground within and adjacent to the field was

TABLE I

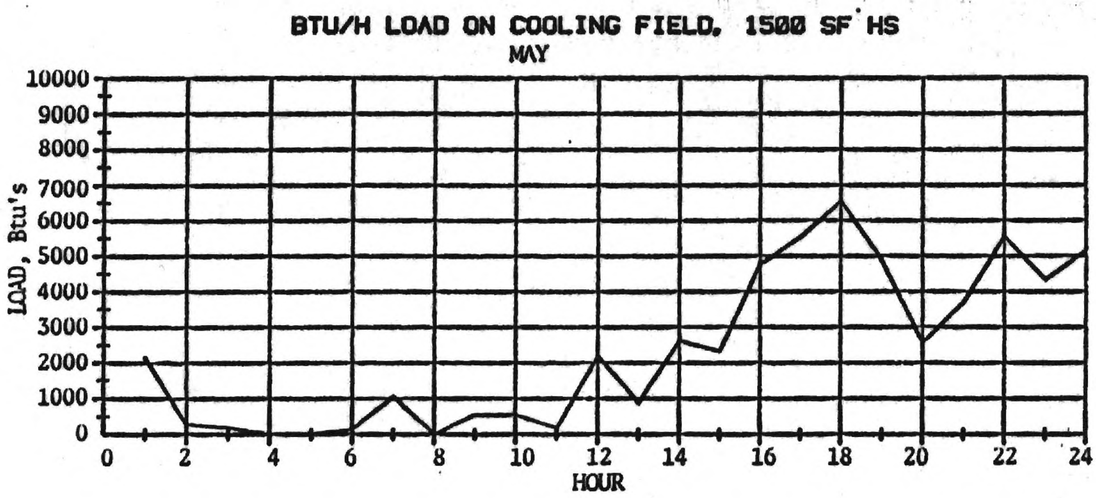
Daily Sensible Cooling Load for Georgia Power Colombus House and Percentage Carried Passively

Month	Cooling Required (BTU/Month)	Cooling Passive (BTU/Month)	Percentage
May	1,280,216	0 *	0
June	2,508,555	1,421,515 **	56.7
July	2,444,049	1,024,920 ***	41.9
Aug.	2,158,381	0	0
Sept.	1,863,498	0	0
TOTAL	10,254,699	2,446,435	23.9

* Passive cooling not used in May due to load simulator failure

** Passive Cooling not started until 12 June due to load simulator failure

*** Cooling capacity of field exhausted on 16 July due to partial charging during previous winter



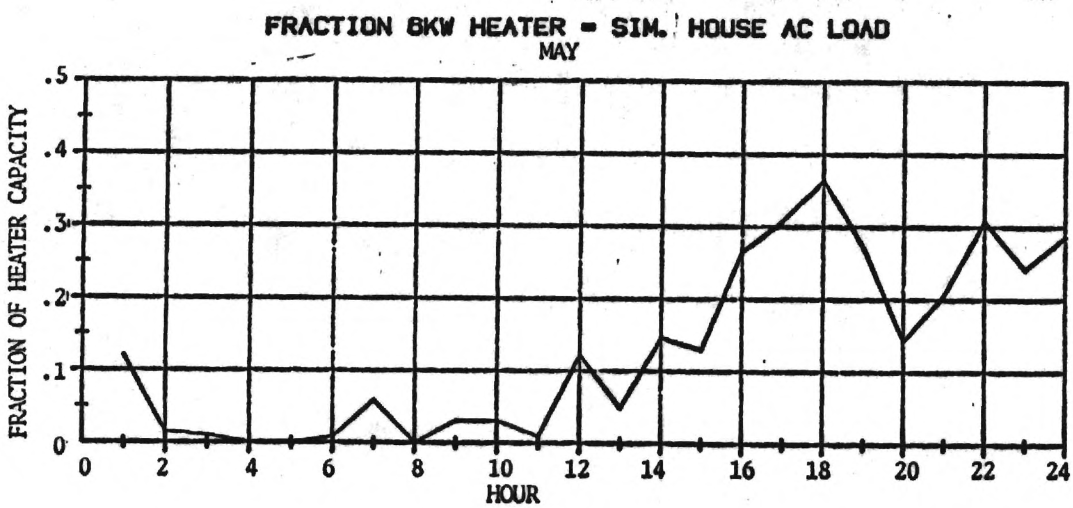
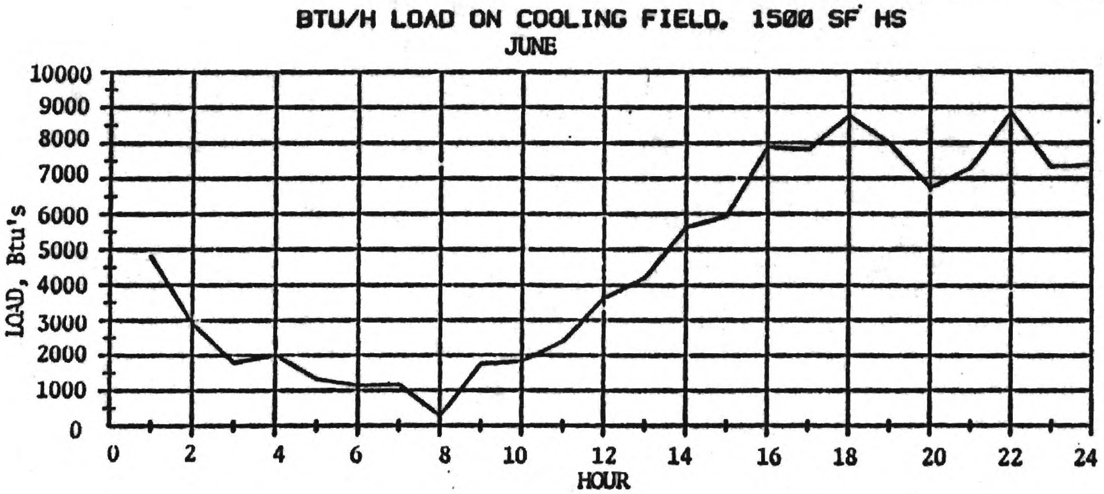


Figure 1 Building load simulation profile for May



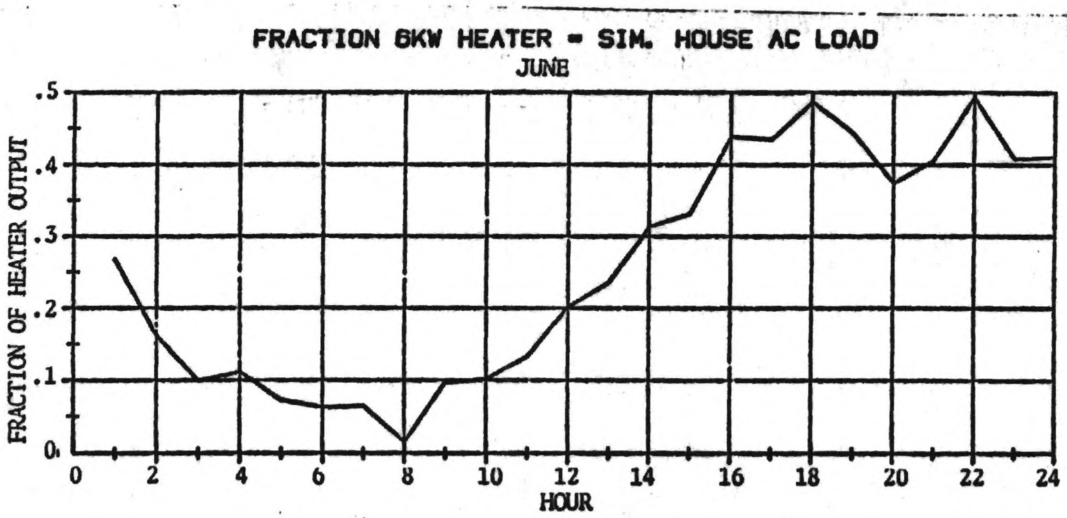
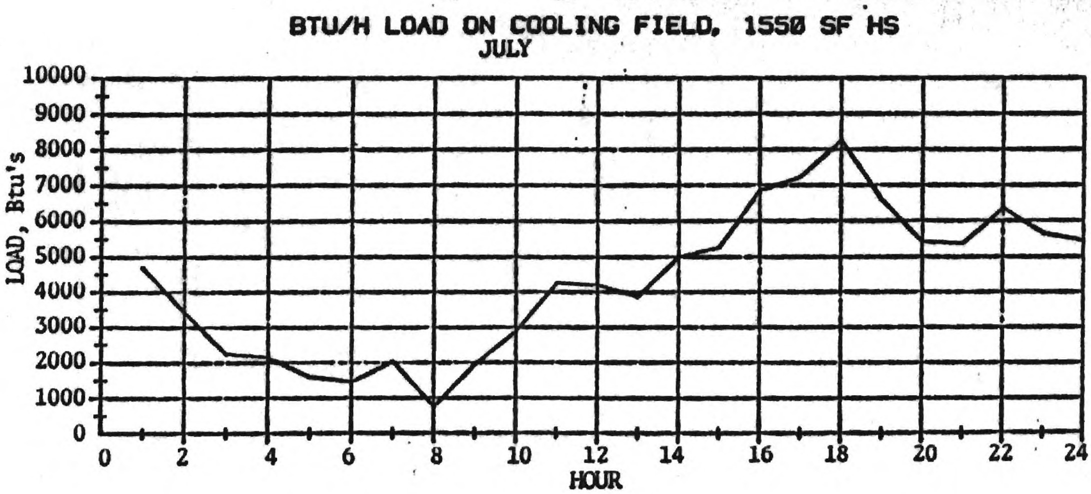


Figure 2. Building load simulation profile for June



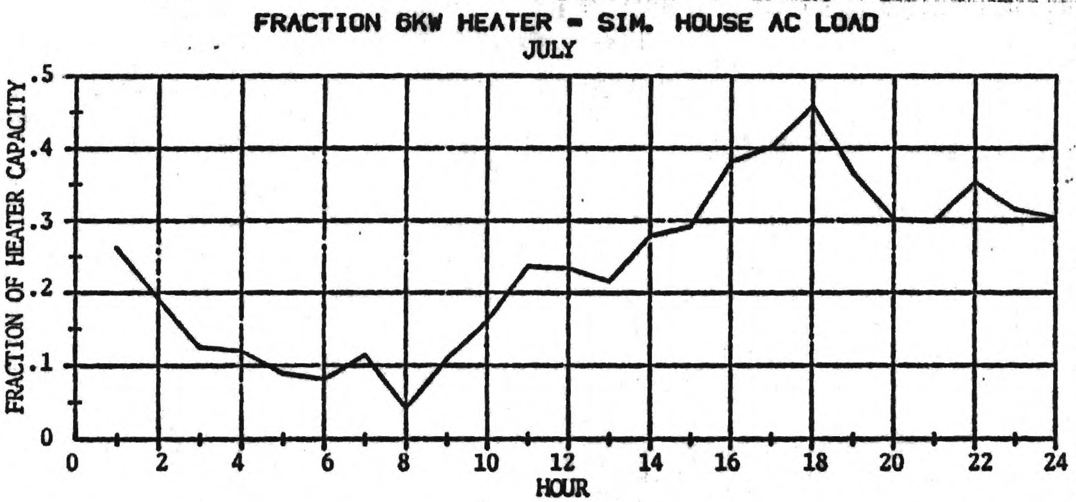
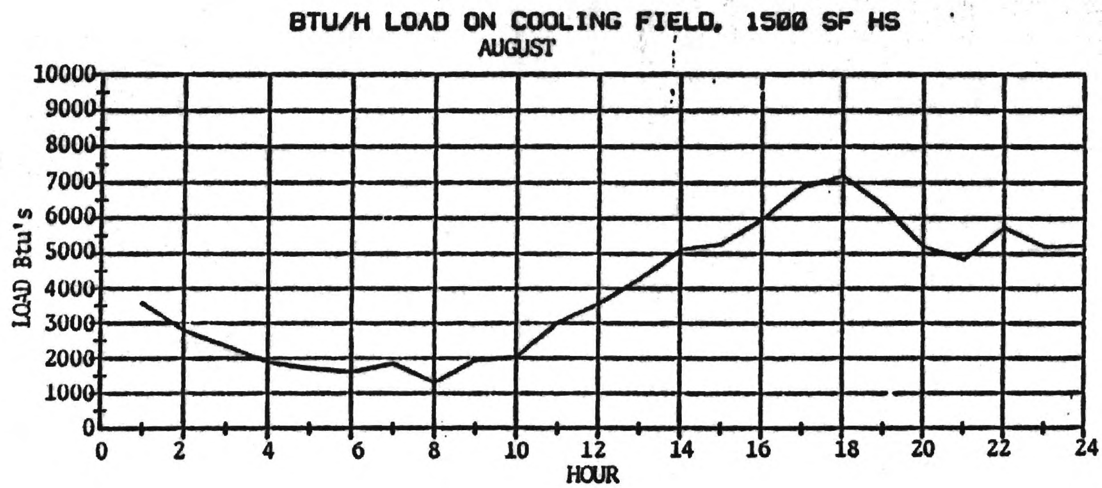


Figure 3. Building load simulation profile for July



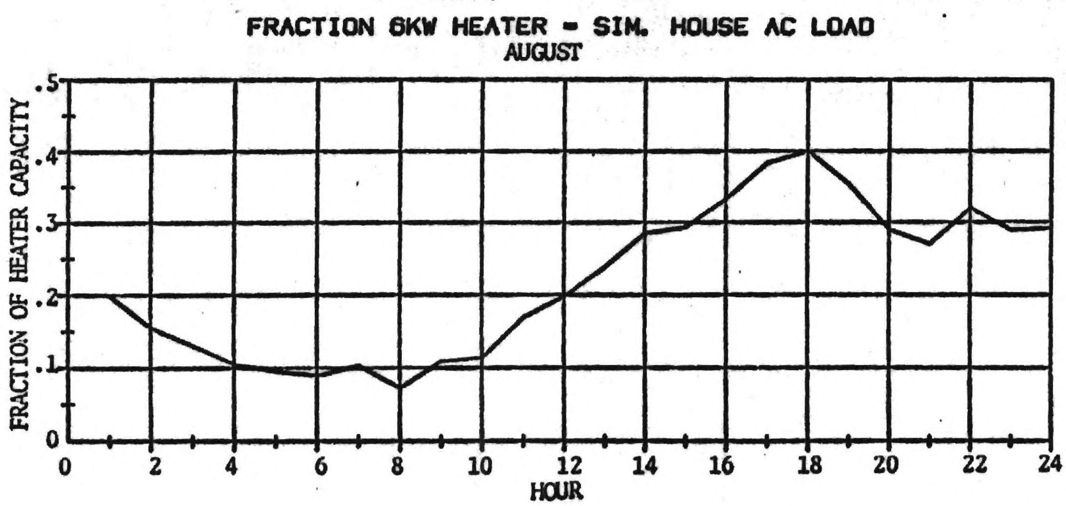
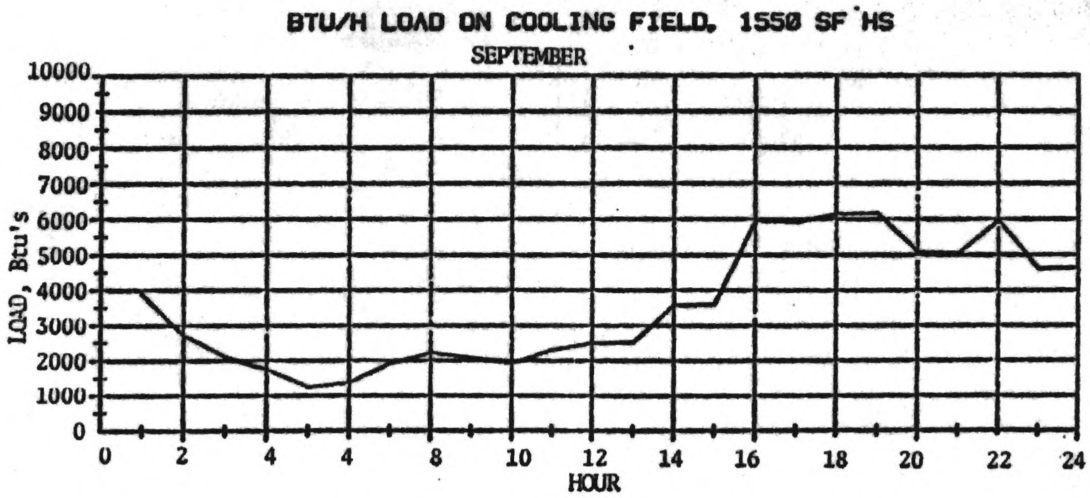


Figure 4. Building load simulation profile for August



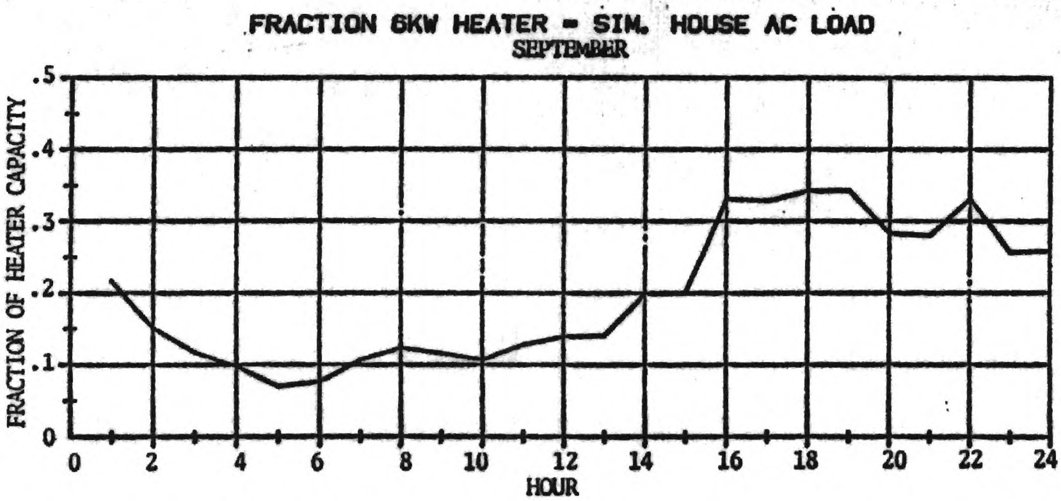


Figure 5. Building load simulation profile for September

much warmer than the ground at similar depths which had not been subjected to the conditions described above. The insulation minimized the loss of energy to the surface. Because of the high initial temperature, the insulation and the late start in charging; the field was not cooled to the temperature it would reach during a normal winter season changing cycle.

The field was scheduled to start carrying the simulated load in May. Initial checkout showed all of the equipment functioning as desired but when the load sequence was attempted in May, it failed to work. The problem was finally traced to a failure in the process controller. This controller is normally very reliable. The controller required off-site repair and was not available for use until the end of the second week in June.

Since the cooling field performs like any energy storage system with imperfect insulation, the field was losing cooling capability during the six week delay caused by the controller failure. Had the system been operable at the beginning of May, it would have passively carried all of the May and June cooling loads despite the severe handicaps of the high initial temperatures and late initiation of charging. The system performance during the past year, despite all of the problems, leads one to be very optimistic about the potential performance of the DET concept.

Radiative Cooling

Considerable effort had been directed toward establishing the performance of both concrete and steel radiative cooling panels under both steady state and transient load conditions. Figure 6 shows a schematic of the test apparatus which was used to control and maintain a preset water temperature flowing into the test panel. Figure 7 shows a schematic of the programmable load control for the test box. Figure 8 is a photo of the test box while open and Figure 9 shows a 2 inch (half thickness) test panel being readied for positioning in the test box.

Computer simulations using the GROCS program have shown that while the cooling field will have sufficient cooling capacity to meet the average daily load of a well designed residence, it might have difficulty meeting the instantaneous peak load, especially after operation for a month or more. This results from the soil immediately adjacent to the pipe being heated more rapidly than the energy can diffuse through the low conductivity soil. While one could get around the problem by going to smaller diameter pipe in the field with close spacing and greater length, a more practical solution appears to be through the addition of mass to the radiative cooling panels.

Computer simulation of the performance of the radiature cooling panels using both MITAS and TNOOE showed that pipes within the wall might have to be placed as close as 4" apart if one has a steady state load. The simulations also showed that under transient loads the mass of a radiature cooling wall stores sufficient cooling capacity, if properly sized, that temperature gradients between tubes spaced 10-16" apart may be satisfactory. Although insufficient experimenmtal data are available to arrive at conclusions at this time, it does appear that pipe spacing of 8-10 inches are practical.

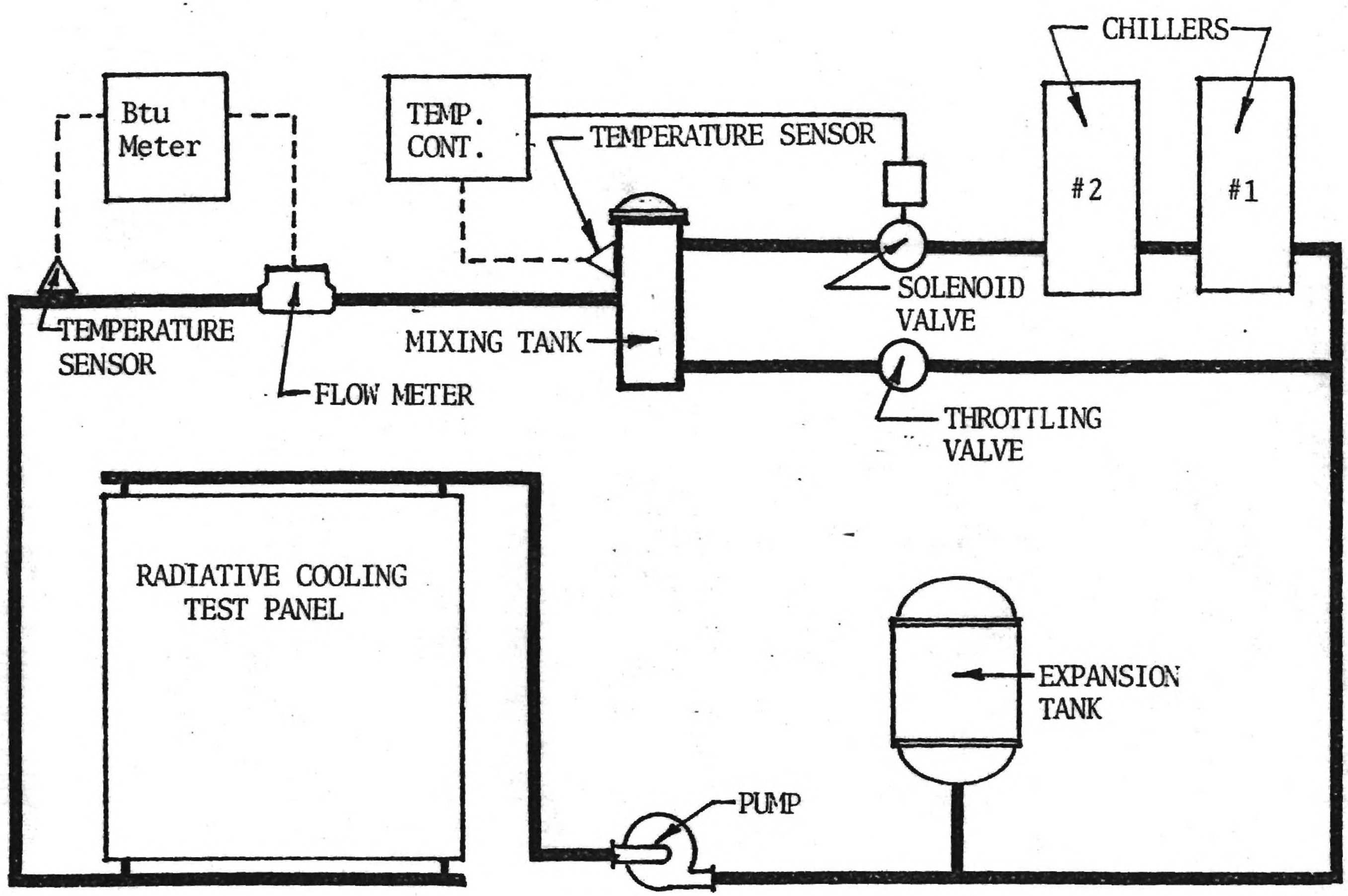


Figure 6. Schematic of test panel water and temperature control

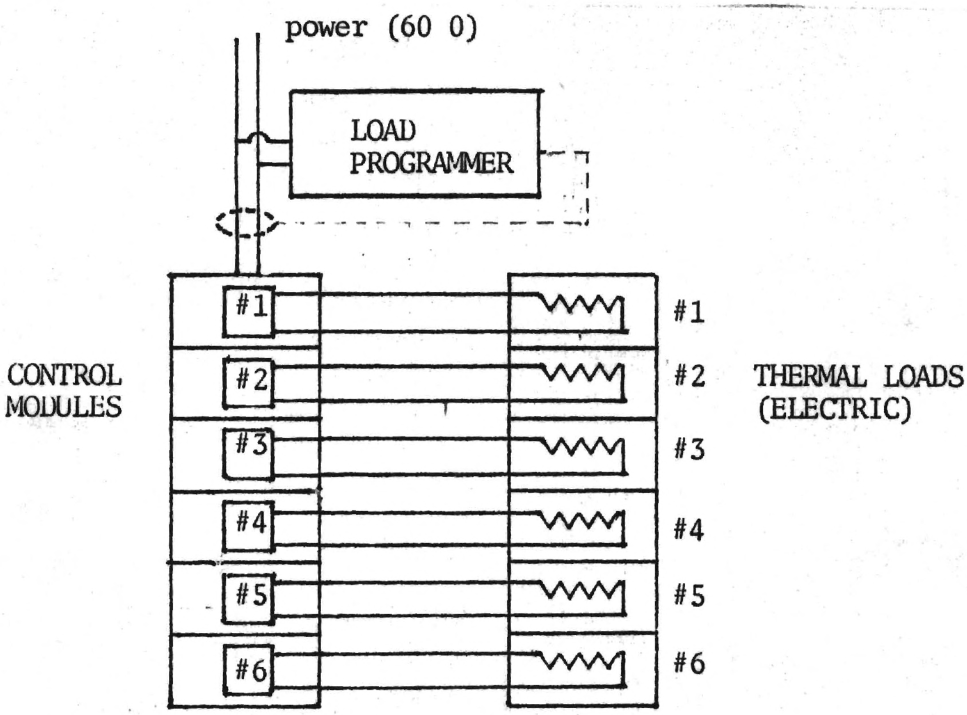


Figure 7. Schematic of programmable load control for test box

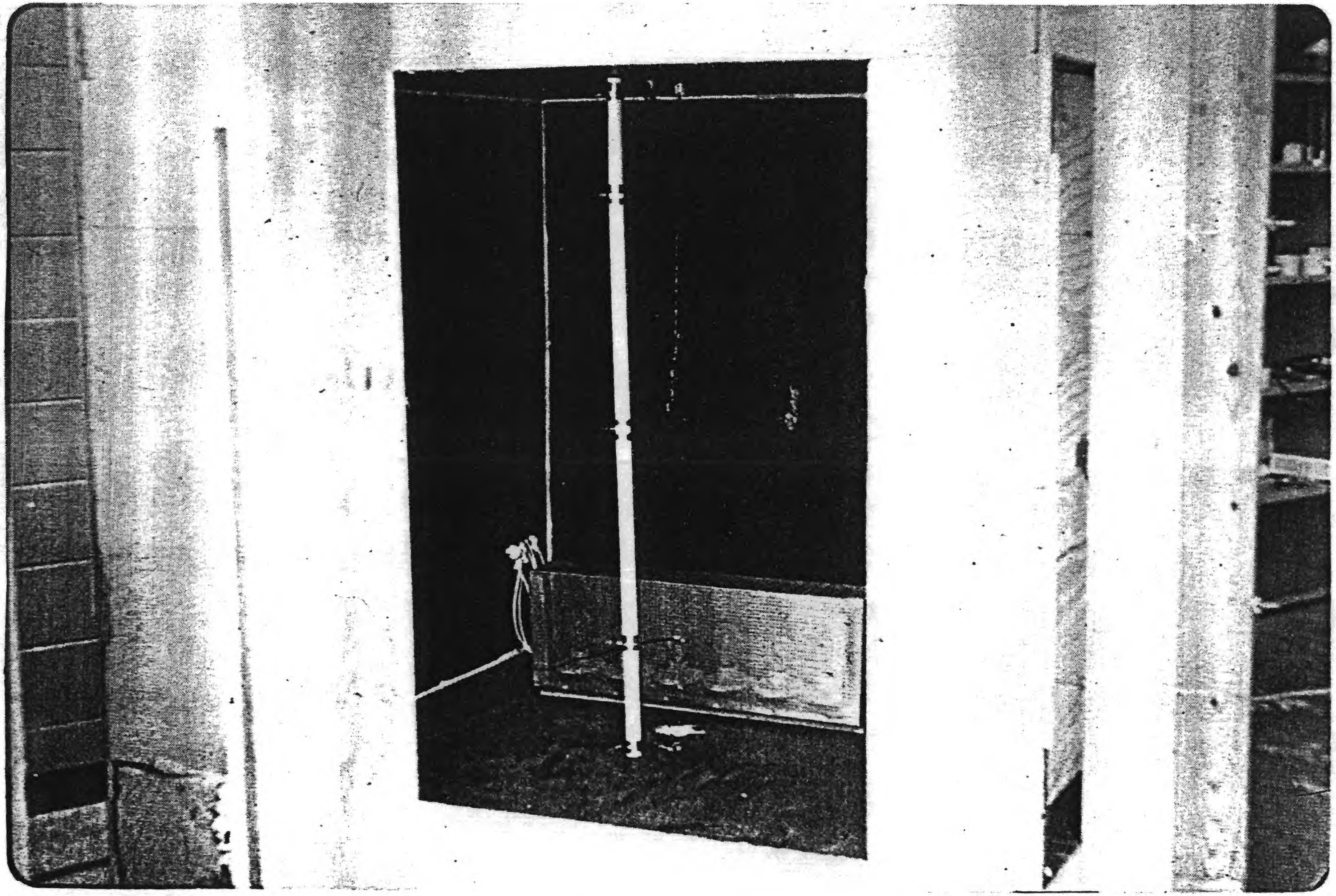


Figure 8 View of open test box

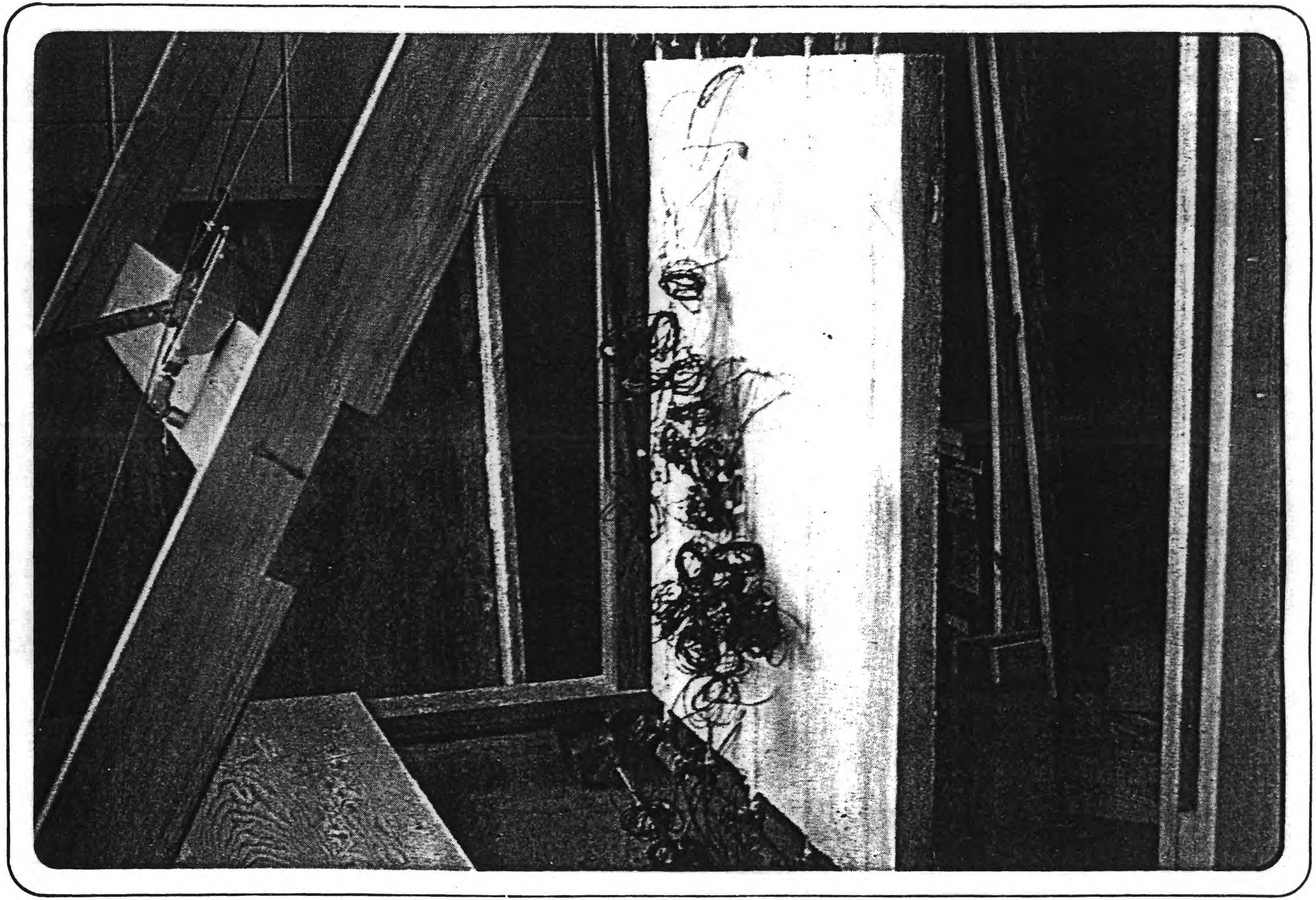


Figure 9. View of 2" test panel before installation

It was felt from the beginning of this program that it was imperative that any proposed passive cooling system work well, or at least not interfere, with potential passive heating systems. It is also important that both passive heating and cooling systems be complemented with efficient auxilliary heating and cooling systems. Nothing useful is accomplished if much of the energy one saves with passive heating and/or cooling systems is lost through the use of inefficient auxilliary systems. Unfortunately many advocates of passive systems are opposed to incorporation of state-of-the-art or high technology mechanical systems as a backup. This usually results in poor efficiency and less comfort during periods requiring mechanical conditioning.

Considerable effort has been directed toward investigation of both compatible high efficiency auxillary mechanical systems and compatible high efficiency dehumidification systems. The incorporation of an efficient system to carry latent loads is mandatory. As reported in PR No. 1, feasible passive techniques are simply not available.

It was felt from the start of this program that it would be highly unlikely that a passive cooling system for hot-humid climates could be developed that would be capable of meeting 100% of a residential sensible cooling load. This meant that an auxilliary cooling system is necessary if comfort is to be maintained. It was also felt that passive techniques for meeting latent cooling loads are not likely to be developed in the near future. If the sensible cooling load of a building is to be met radiatively in a humid climate, it is imperative that latent loads be efficiently handled. This means that auxilliary systems are needed to handle latent only at times and sensible and latent during extreme periods.

Unfortunately, if one were to employ a conventional air conditioner to handle the latent load, it would also provide sensible cooling which could be provided passively. It appears that greater efficiency can be obtained by handling the sensible and latent loads with separate equipment rather than with a single component as is normal practice.

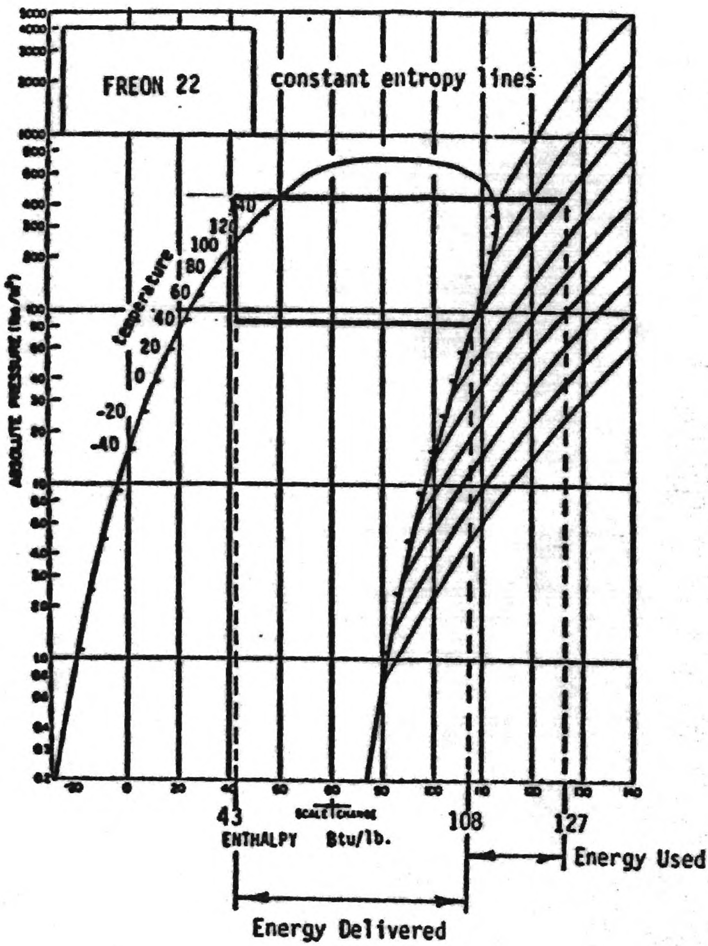
Auxillary Sensible Load

If one plots an idealized Rankine cycle air source air conditioner on a pressure enthalpy diagram one would have a cycle such as shown in Figure 10. The coefficient of performance (COP) would be about 3.42 for the most efficient systems presently available. One is limited to this COP by two factors. First, due to high ambient temperatures, one must have condenser temperatures of 150°F or above to dissipate the energy removed from the residence to the ambient air. One also must have evaporator temperatures at or below 50°F if one is to adequately handle the latent load.

Use of a conventional air-source heat pump would not meet our stated desire to handle the sensible and latent loads separately. One can meet the sensible cooling load passively until the temperature of the water coming from the cooling field reaches 72-75°F. If one now supplies the 72-75°F water coming from the field to the condenser of a water-to-water heat pump and supplies water from the heat pump evaporator to the cooling walls, one can function with a Rankine cycle similar to the one shown in Figure 11. Notice it is not now necessary to operate the condenser at 150°F because of the 72-75°F water available from the field. It is also not necessary to operate the evaporator at 50°F because the radiative cooling wall works well with 70°F water. One now has a auxilliary sensible cooling system with a COP of 6.0. Figure 12 gives a schematic of the cooling field and cooling wall when operating through the water-to-water heat pump.

CONVENTIONAL AIR-SOURCE HEAT PUMP : COOLING

PRESSURE-ENTHALPY DIAGRAM

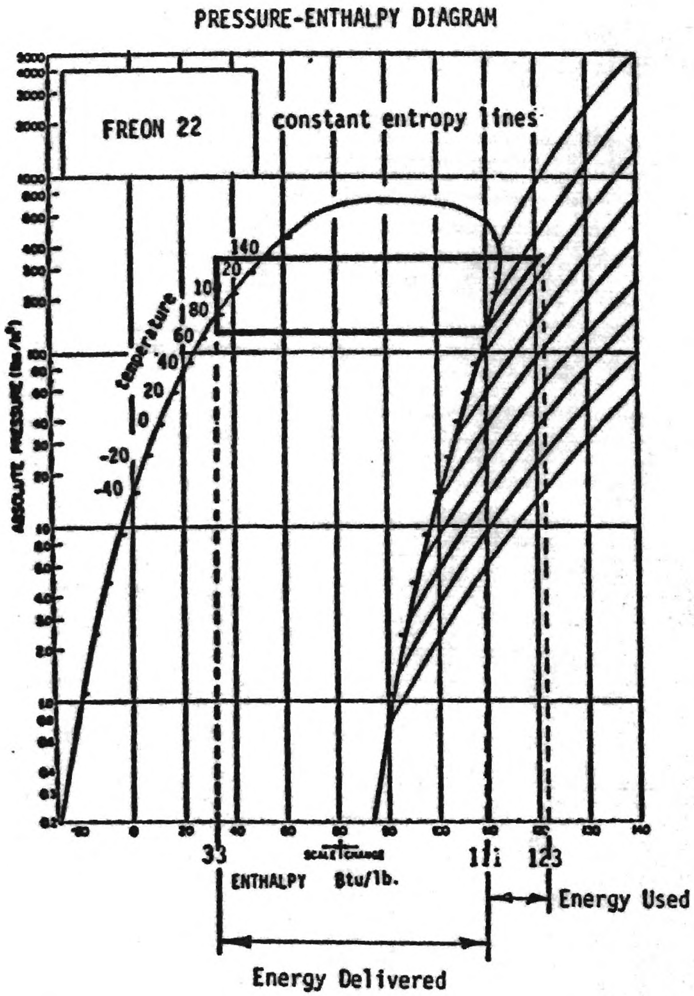


$$COP = \frac{\text{ENERGY DELIVERED}}{\text{ENERGY USED}}$$

$$COP = \frac{108-43}{127-108} = 3.42$$

Figure 10 conventional heat pump-cooling

WATER-TO-WATER HEAT PUMP : COOLING



$$COP = \frac{\text{ENERGY DELIVERED}}{\text{ENERGY USED}}$$

$$COP = \frac{111-33}{124-111} = 6.0$$

Figure 11 water-to-water heat pump-cooling

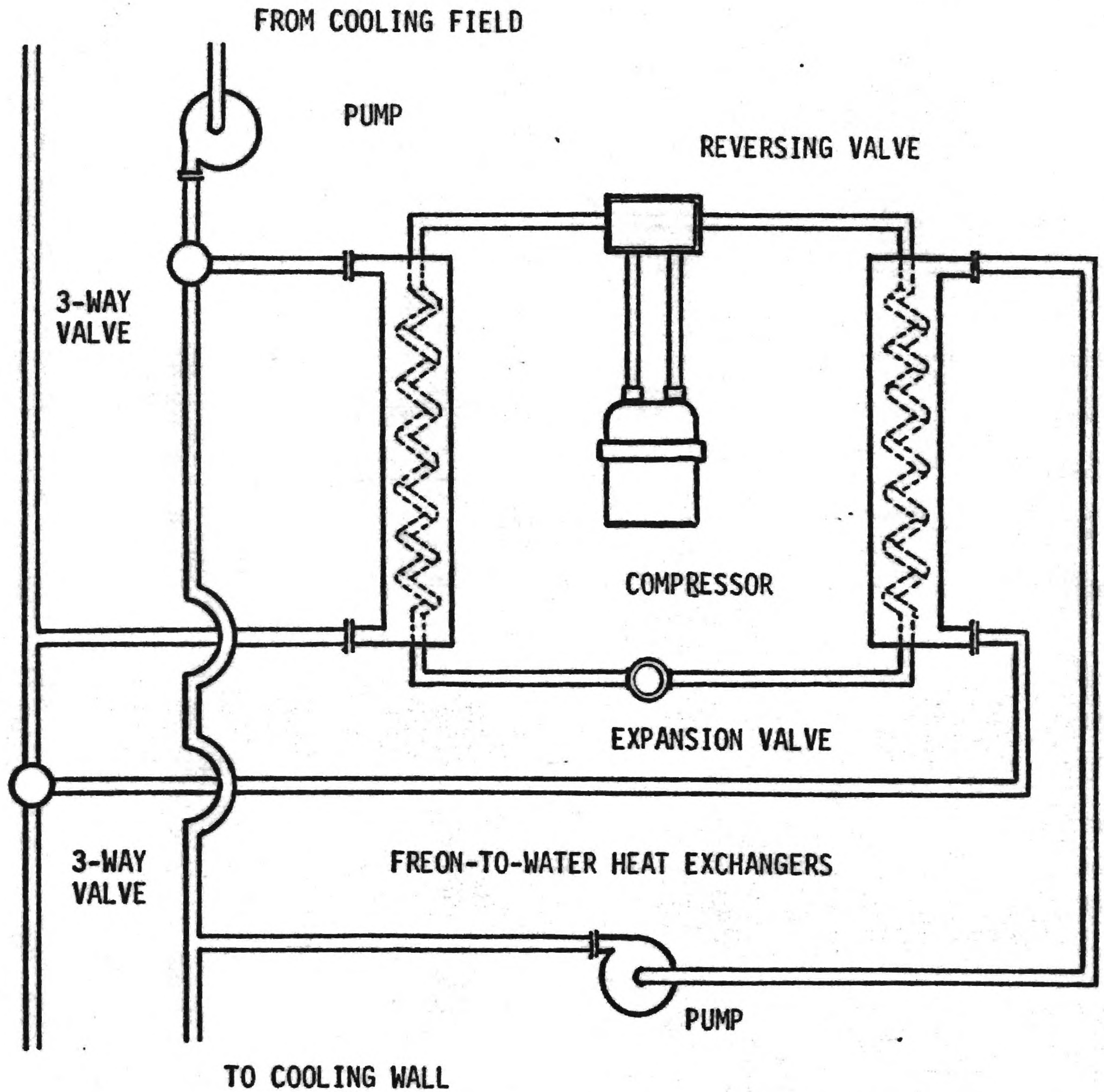


Figure 12 Passive/active cooling system

Similar cycles can be shown for a conventional heat pump and a water-to-water heat pump in the heating mode. One finds the COP improves from 3.10 to 6.92 by going to a water-to-water heat pump. This system obviously meets our requirement for a high efficiency sensible auxilliary system which is compatible with the passive system.

Auxilliary Latent Load

If one succeeds in passively heating and cooling a residence 100%, one finds that a substantial energy requirement still remains. It is not uncommon for domestic hot water (DHW) energy requirements to exceed the heating and cooling requirements for well designed energy efficient conventional homes. Several manufacturers have recently marketed domestic hot water heaters which operate on a heat pump principle. These heat pump DHW heaters require only 40-50% as much energy input as conventional electric resistance DHW heaters. If one locates the heat pump DHW heater in occupied space it not only heats the domestic hot water more efficiently, it also provides sensible and latent cooling. Figure 13 shows a heat pump DHW heater modified with a run-around coil. The run-around coil decreases the sensible cooling capacity and increases the latent cooling capacity without changing the total capacity or significantly affecting the efficiency of the heat pump as a DHW heater.

We now have a efficient DHW heater, a very efficient sensible auxilliiary system and a latent auxilliary system which is a by-product of the DHW heater.

Other Work

Work during this report period has also involved preparation for and attendence at two D.O.E. meetings. This required the preparation of two papers. Additionally a paper has been prepared for the ISES Passive Conference in Portland, Oregon. Two papers related to this program have also been prepared for presentation at the ISES Passive/Hybrid Cooling Conference in Miami. Copies of these papers are included as appendices A-E.

Problems

In addition to the various control and scheduling problems discussed here and in earlier reports, we have just encountered a series of leaks within the cooling field requiring portions of the field to be dug up and repairs made. At this time it appears the leak problems are related to failure to screen the backfill for rocks and carelessness on the part of the bulldozer driver when covering the field tubing. All leaks have been traced to cracks caused by large rocks or by the dozer track.

Future Work

Work during the next report period will concentrate on completion of construction of the large test room, conversion of the heat pump to a water-to-water mode and modification of a heat pump domestic hot water heater to include a run-around coil, readying the field for the coming season's charging, and continued testing of potential building side heat exchange strategies in the calormetric box.

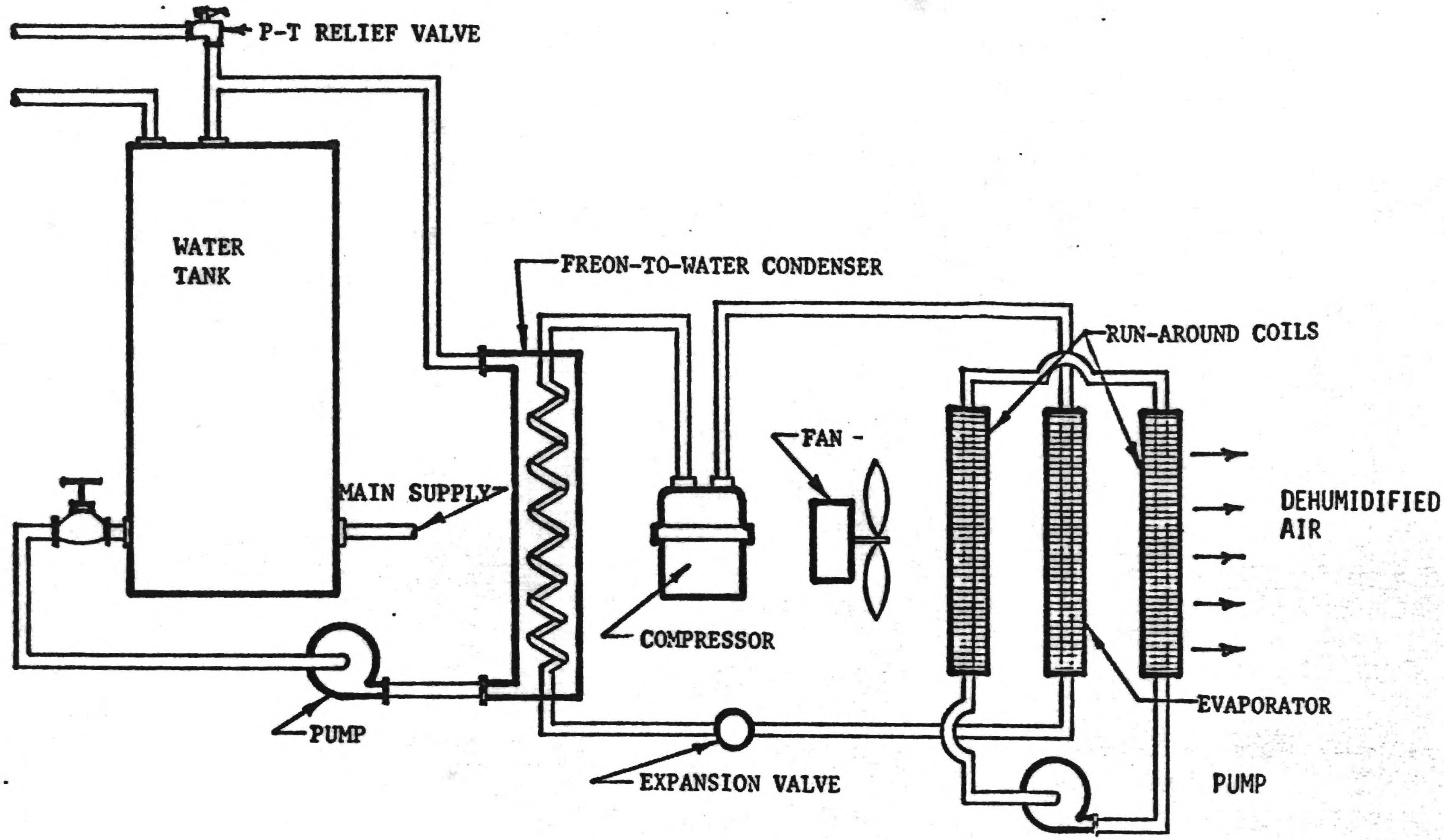


Figure 13 Modified heat pump DHW heater

APPENDIX A

"Passive Cooling For Hot-Humid Climates"

Passive Cooling Program Mid-Year Contractor
Review Meeting: June 22-26, 1981

PASSIVE COOLING FOR HOT HUMID CLIMATES

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ABSTRACT

Earth Tempering is a term presently used for structures which are either buried or semi-buried. These structures typically use the ground as an insulating element and as a high capacitance barrier for the dynamic attenuation of heat flux.

In the proper environmental context, an earth tempered building can perform quite well. Unfortunately, underground buildings can be expected to meet considerable market resistance due to a number of factors. These include an increase in construction costs, waterproofing criteria, excessive site modifications and mass public resistance to subterranean housing. This has led to a possible cooling concept which the authors call "Detached Earth Tempering" (DET). The scheme involves an intimate but indirect coupling of selected building elements to the earth as a heat sink. Successfully applied, the concept promises the advantages of below-ground earth tempered construction without the structural, moisture, site and cost liabilities usually associated with below-grade buildings. This paper addresses the advantages and disadvantages of Detached Earth Tempering and describes a research program which is currently investigating the concept. The paper presents data on measured performance and predicted performance.

1. INTRODUCTION

Passive cooling in hot-humid climates has proven to be one of the most difficult challenges for natural building systems. Numerous literature searches have revealed little, either in terms of architecture or devices, anywhere in the world which shows promise as a passive cooling method for hot-humid climates. The most prevalent and effective strategy is not a cooling concept but a load minimization technique. If air is already hot and humid, it is imperative that radiant loads due to the sun be minimized or eliminated. Shading devices are prevalent and extensive throughout the hot-humid regions of the world.

Our study did show that earth tempering, in the form of underground construction, could significantly reduce sensible cooling loads in hot-humid climates if air infiltration could be controlled. Despite this potential, few examples of earth tempering in hot-humid climates exist. Unlike most passive techniques, earth tempering in hot-humid climates has been held back by a technical problem. Until recently, it was not possible to control infiltration sufficiently well to make earth tempering practical in hot-humid climates.

The importance of infiltration or ventilation control in hot-humid climates becomes very apparent when one looks at the latent and sensible loads of a building as a function of the ventilation (infiltration) rates while keeping the ventilation air temperature constant. If the relative humidity of the air is now varied, one finds the only difference in the thermal load on buildings in arid and humid climates is due to the latent loads caused by ventilation (infiltration). Obviously, if infiltration can be greatly reduced and carefully controlled, a building in a humid climate will perform nearly the same as in an arid climate.

Although the study showed heating and cooling potential for earth tempering, it also showed serious architectural and market constraints on conventional earth tempered (underground) buildings. The study also showed that many of the thermal advantages of underground construction may be realized above ground through the use of a concept we have chosen to call "Detached Earth Tempering" (DET). If architectural constraints prevent taking the building underground, the Detached Earth Tempering Concept attempts to bring the thermal advantages of underground structures to above grade buildings.

2. BASIC CONCEPT

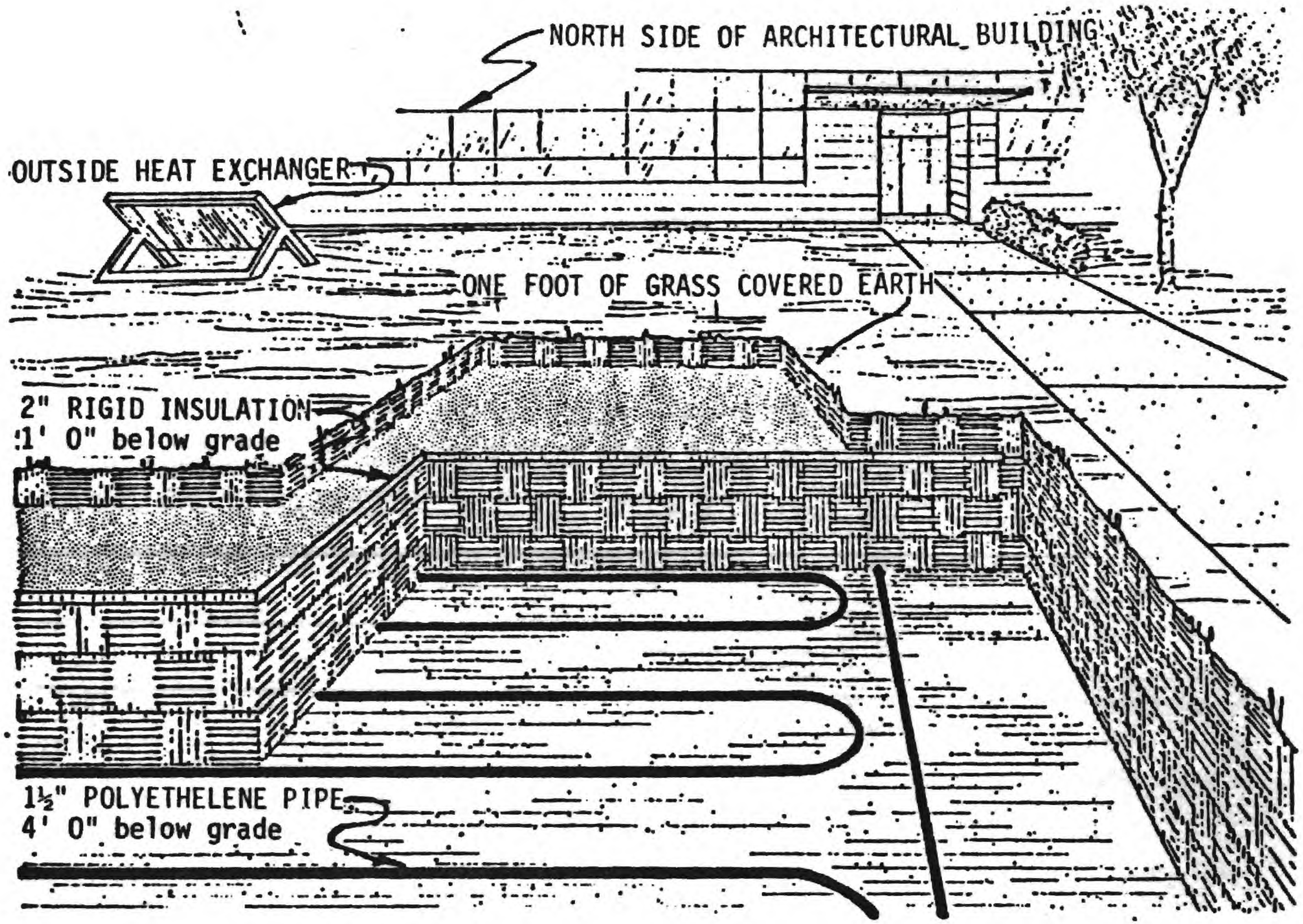
The basic concept behind Detached Earth Tempering is to bury coils in the earth through which water or other similar heat transfer fluids can circulate. The fluid having been cooled by earth contact can then be circulated through building elements such as the floor, ceiling or walls. If the walls are well insulated and the insulation is located on the outside of the structure, one will have a cool wall structure similar to that of an underground structure. If infiltration is controlled through the use of good seals, vapor barriers and air locks at the doors and ventilation is accomplished through the use of an enthalpy exchanger, the building will perform similar to an underground structure in an arid climate.

Initial computer studies showed that ground temperatures at depths of 1.2--3.6 meters (4--12 feet) are much too high in late summer to provide appreciable cooling. The computer studies were checked with ground temperature measurements. These measurements verified the predicted ground temperatures for areas well shaded. They also showed that areas with little ground cover can reach considerably higher temperatures.

Because high surface temperatures result in high temperatures at greater depths, one might minimize this effect by separating the surface from the lower depths with insulation. Insulation of the soil from the surface also greatly reduces the rate at which energy can be lost to the ambient air during the winter months. This then requires that the soil beneath the insulation be cooled during the winter months if one is to have the low soil temperatures desired during the summer months.

Georgia Tech has installed an experimental field with 213 meters (700 feet) of 38 mm (1.5 inch) polyethylene pipe buried at a depth of 1.2 meters (4 feet) with .9 meters (3 feet) of dirt directly above, followed by 51 mm (2 inches) of extruded polystyrene insulation. The insulation is covered with .3 meters (1 foot) of dirt with a good sod cover. Ideally the field would be placed beneath the house to minimize undesirable ambient loads. Figure 1 shows a section of the Georgia Tech experimental field. Since it could not be installed beneath a house, great care has been exercised in providing a good sod cover to minimize radiant gains at the soil surface.

The insulated field is cooled during the winter months by circulating water through an above ground air-to-water heat exchanger and then through the buried coil. A differential thermostat turns on a small (1/15 hp) pump when the field is warmer than the ambient air.



COOLING FIELD SECTION

Figure 1 Field cross-section

The success of seasonal storage of cooling potential is highly dependent upon how one couples this cooling capacity to the occupants. Due to the low grade (temperature differences are relatively small) cooling potential, conventional cool air systems will not perform satisfactorily. Cooling through the use of building elements such as walls, floors or ceilings appears to offer the most potential. These elements provide large heat exchange surfaces in direct radiant contact with building occupants.

Initial computer studies showed that cooling capacity from a buried field might be incapable of meeting peak instantaneous loads but would be adequate for average daily loads. This indicated that optimum performance could not be obtained using low mass radiant planes. Although considerable data have been published in the literature directed toward the design of radiatively heated buildings, there are little data on design of radiatively cooled buildings. ASHRAE[1] provides some design guidelines using low mass radiatively cooling panels.

Radiative cooling potential of concrete walls of several thicknesses with several different tube spacings have been simulated using a thermal simulation program called MITAS[2] and a smaller thermal network program for microcomputers called T-NODE[3]. These simulations show the radiative cooling concept to have potential. The simulations have also shown the need for experimental data on the performance of such walls due to uncertainties about convective heat transfer coefficients on cooled walls.

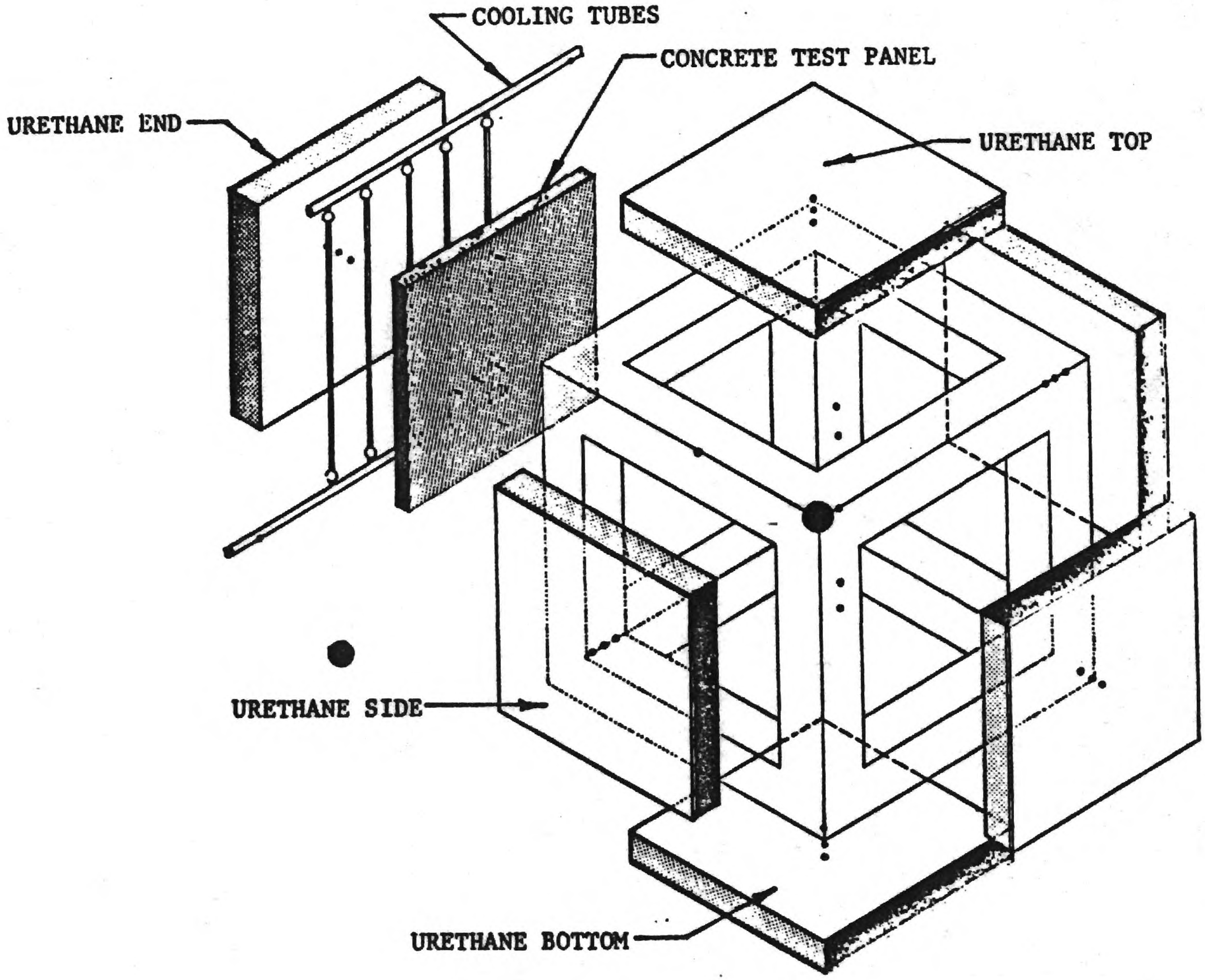
A radiative panel test chamber has been designed, constructed and is currently being used to develop experimental data on the performance of cooling walls. Figure 2 is an exploded view of the radiative cooling test chamber. This chamber has the capability of quantifying the cooling performance of walls, floor and ceiling elements.

3. SYSTEM PERFORMANCE

Due to completion of the field installation and control system late last fall, combined with the high ground temperatures trapped beneath the insulation when the insulation was installed in late September, the field temperatures were not dropped to the levels last winter that they would reach during a normal cycle. Figure 3 shows temperatures across the field at a four foot depth for several dates. Notice on March 15th that the ground temperatures adjacent to the field (thermocouples number 14 and 41) are approximately the same as those within the field. This would not be expected when the field has run through a complete cycle.

Figure 4 shows temperatures along a vertical shaft at the center of the field at the same time as the data given in Figure 3. Note that temperatures at depths below the four foot level increase with depth. Notice that the field temperature on May 29 in figure 4 is well below the adjacent ground temperature but is substantially above its temperature on March 15. At first this rapid rise in temperature was quite disturbing until one realized the tremendous sink below the field that was at a temperature well above the field temperature on March 15. This is shown quite clearly in Figure 4. Although some energy is diffusing from the surface through the insulation and warming the field, the field is also cooling the block of earth beneath it.

The field was scheduled to begin cooling a simulated building, using a resistance heater and load programmer, in early May. The load programmer failed during the initial checkout and couldn't be replaced until June 12 causing the load simulation tests to be delayed until that date.



EXPLODED VIEW OF RADIATIVE PANEL TEST BOX

Figure 2 Cooling panel test box

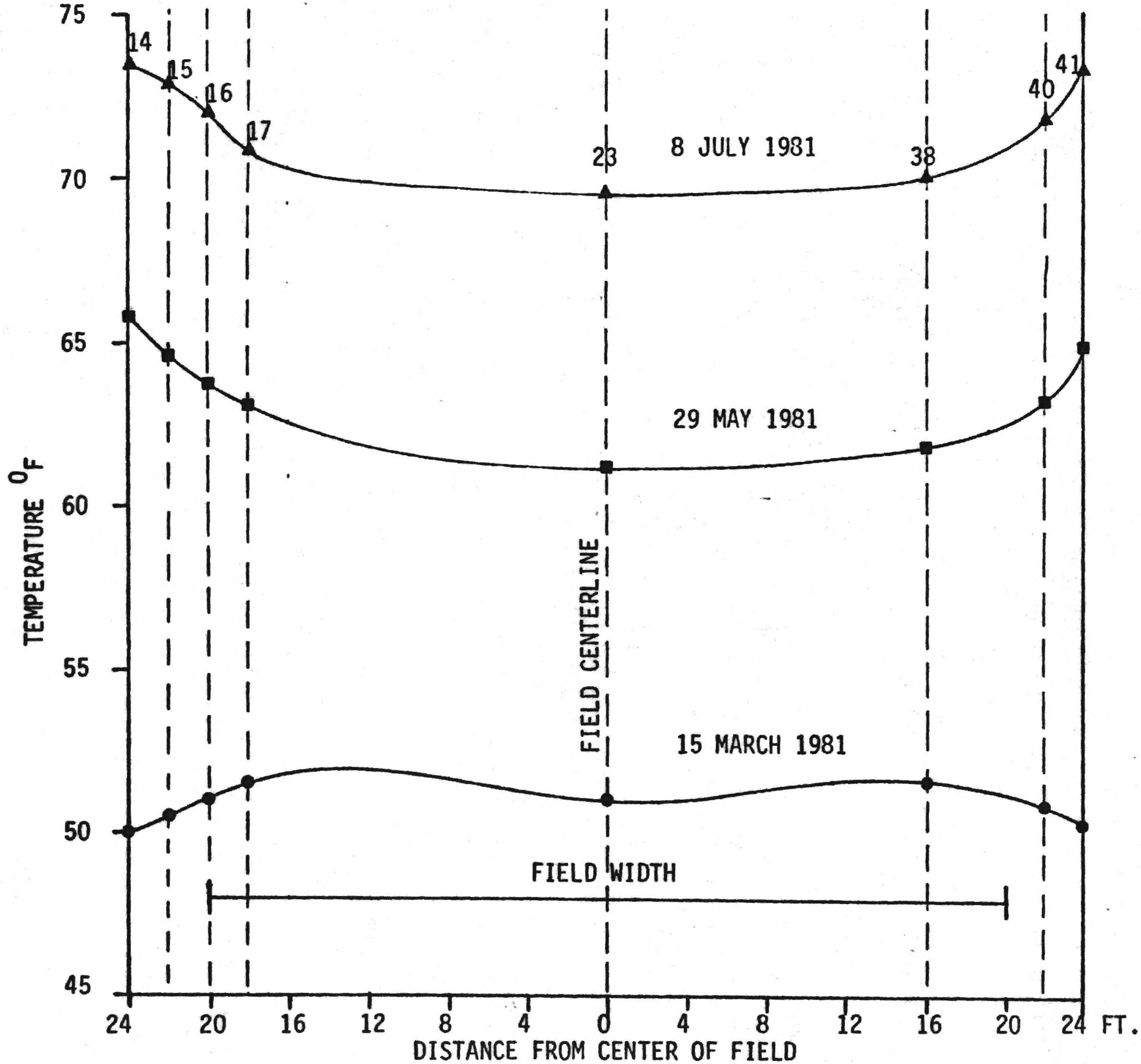


Figure 3 Temperatures across the field

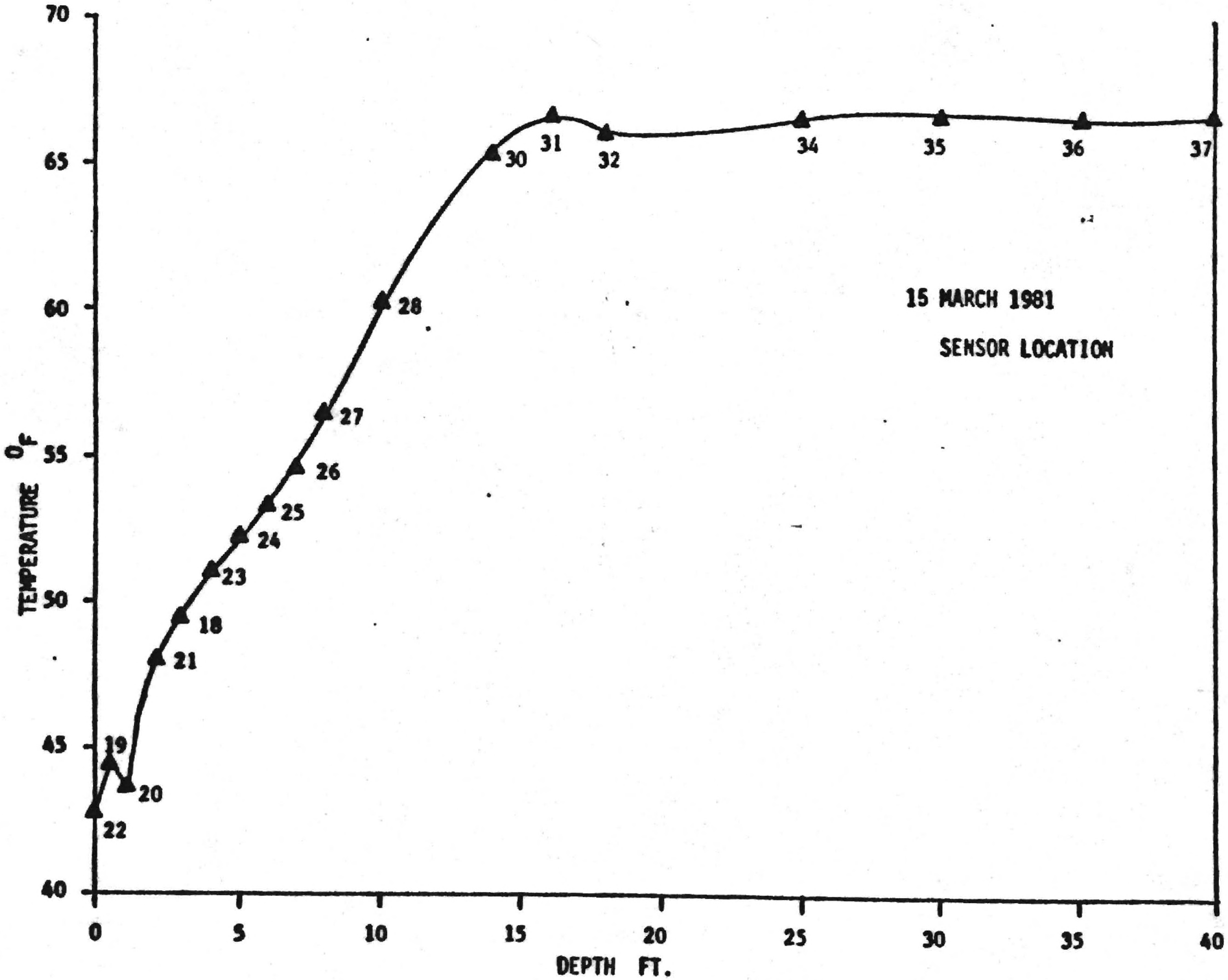


Figure 4 Field temperature vs depth

The measured load for an existing energy efficient, but conventional design, home located in Columbus, Georgia was programmed in the simulator. The field has been carrying the entire sensible load of the building except for several days when the circulating pump lost prime and a leak in the field caused a brief shutdown. The July 8 curve on Figure 3 shows the field temperature is beginning to approach the point where it will be unable to provide water at a temperature low enough to radiatively cool. This early depletion of cooling potential was expected due to the late cooling start last winter and the high ground temperature resulting from the insulation being installed when the ground was the hottest.

4. AUXILLIARY SYSTEMS

It is imperative that passive cooling systems work well, or at least not interfere, with passive heating systems. It is also important that both passive heating and cooling systems be complemented with efficient auxilliary heating and cooling systems. Nothing useful is accomplished if much of the energy one saves with passive heating and/or cooling systems is lost through the use of inefficient auxilliary systems. Unfortunately many advocates of passive systems are opposed to incorporation of state-of-the art or high technology mechanical systems as a backup. This usually results in poor efficiency and less comfort.

It was felt from the start of this program that it would be highly unlikely that a passive cooling system for hot-humid climates could be developed that would be capable of meeting 100% of a residential cooling load. This meant that an auxilliary cooling system was necessary if comfort was to be maintained. It was also felt that passive techniques for meeting latent cooling loads are not likely to be developed in the near future. If the sensible cooling load of a building is to be met radiatively in a humid climate it is imperative that latent loads be efficiently handled. This means that auxilliary systems are needed to handle latent only at times and sensible and latent during extreme periods.

Unfortunately, if one were to employ a conventional air conditioner to handle the latent load, it would also provide sensible cooling which could be provided passively. It appears that greater efficiency can be obtained by handling the sensible and latent loads with separate equipment rather than with a single component as is normal practice.

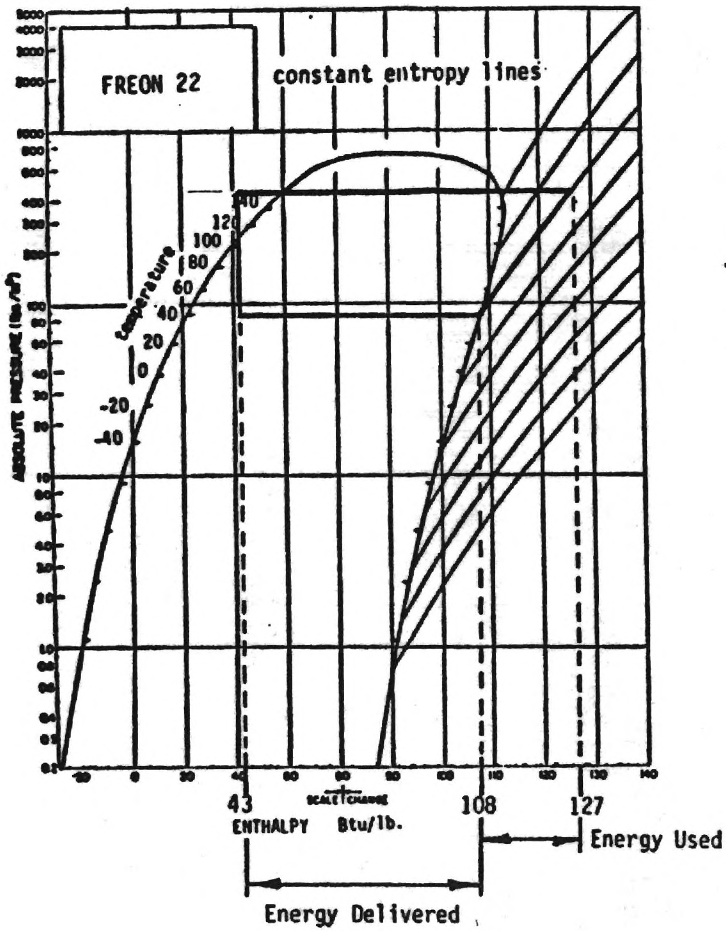
4.1 Auxilliary Sensible Load

If one plots an idealized Rankine cycle air source air conditioner on a pressure enthalpy diagram one would have a cycle such as shown in Figure 5. The coefficient of performance (COP) would be about 3.42 for the most efficient systems presently available. One is limited to this COP by two factors. First, due to high ambient temperatures, one must have condenser temperatures of 150°F or above to dissipate the energy removed from the residence to the ambient air . One also must have evaporator temperatures at or below 50°F if one is to adequately handle the latent load.

Use of a conventional air-source heat pump would not meet our stated desire to handle the sensible and latent loads separately. One can meet the sensible cooling load passively until the temperature of the water coming from the cooling field reaches 74-75°F. If one now supplies the 74-75°F water coming from the field to the condenser of a water-to-water heat pump and supplies water from the heat pump evaporator to the cooling walls, one can function with a Rankine cycle similar to the one shown in Figure

CONVENTIONAL AIR-SOURCE HEAT PUMP : COOLING

PRESSURE-ENTHALPY DIAGRAM

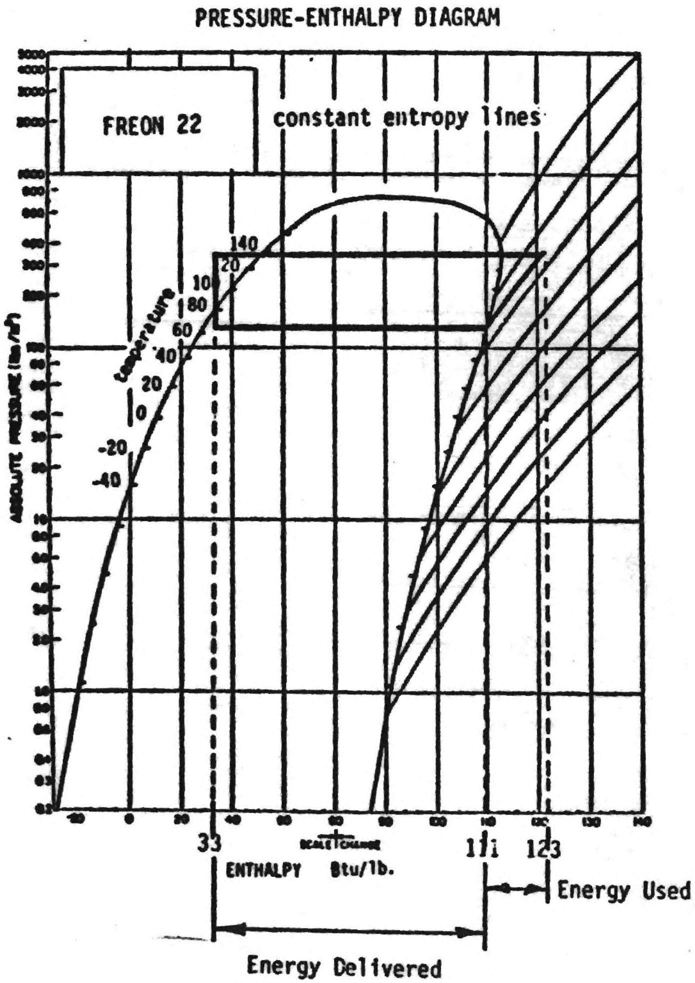


$$COP = \frac{\text{ENERGY DELIVERED}}{\text{ENERGY USED}}$$

$$COP = \frac{108-43}{127-108} = 3.42$$

Figure 5 conventional heat pump-cooling

WATER-TO-WATER HEAT PUMP : COOLING



$$COP = \frac{\text{ENERGY DELIVERED}}{\text{ENERGY USED}}$$

$$COP = \frac{111-33}{124-111} = 6.0$$

Figure 6 water-to-water heat pump-cooling

6. Notice it is not now necessary to operate the condenser at 150°F because of the 74-75°F water available from the field. It is also not necessary to operate the evaporator at 50°F because the radiative cooling wall works well with 70°F water. One now has a auxilliary sensible cooling system with a COP of 6.0. Figure 7 gives a schematic of the cooling field and cooling wall when operating through the water-to-water heat pump.

Similar cycles can be shown for a conventional heat pump and a water-to-water heat pump in the heating mode. One finds the COP improves from 3.10 to 6.92 by going to a water-to-water heat pump. This system obviously meets our requirement for a high efficiency sensible auxilliary system which is compatible with the passive system.

4.2 Auxilliary Latent Load

If one succeeds in passively heating and cooling a residence 100%, one finds that a substantial energy requirement still remains. It is not uncommon for domestic hot water (DHW) energy requirements to exceed the heating and cooling requirements for well designed energy efficient conventional homes. Several manufacturers have recently marketed domestic hot water heaters which operate on a heat pump principle. These heat pump DHW heaters require only 40-50% as much energy input as conventional electric resistance DHW heaters. If one locates the heat pump DHW heater in occupied space it not only heats the domestic hot water more efficiently, it also provides sensible and latent cooling. Figure 8 shows a heat pump DHW heater modified with a run-around coil. The run-around coil decreases the sensible cooling capacity and increases the latent cooling capacity without changing the total capacity or significantly affecting the efficiency of the heat pump as a DHW heater.

We now have an efficient DHW heater, a very efficient sensible auxilliary system and a latent auxilliary system which is a by-product of the DHW heater.

5. ADVANCE MODE OF OPERATION

Once the auxilliary heating and cooling systems have been integrated into the passive design, one finds that a second and possibly better mode of operation becomes possible. One can passively cool with cooling potential stored in a block of earth until the water from the cooling field reaches approximately 74°F. When the water reaches 74°F one actively cools with a water-to-water heat pump using the relatively cool 74°F water from the cooling field. This increases the field temperature until it reaches perhaps 110°F by the end of the summer. One can now passively heat using the 110°F water coming from the field and the radiative cooling/heating wall. When the water coming from the field reaches approximately 85°F, the water is directed through the water-to-water heat pump and the heat pump used to heat through the radiative wall. This cools the field until at the end of the winter the field has been cooled to perhaps 40-50°F. The system is now ready to begin another complete cycle. One now finds that the air-water heat exchanger described earlier and shown in figure 1 is not needed under the new operating mode.

Obviously the cycle will not operate exactly as described due to energy diffusion during the spring and fall. Energy diffusion only changes the temperatures given and not the validity of the proposed operating mode.

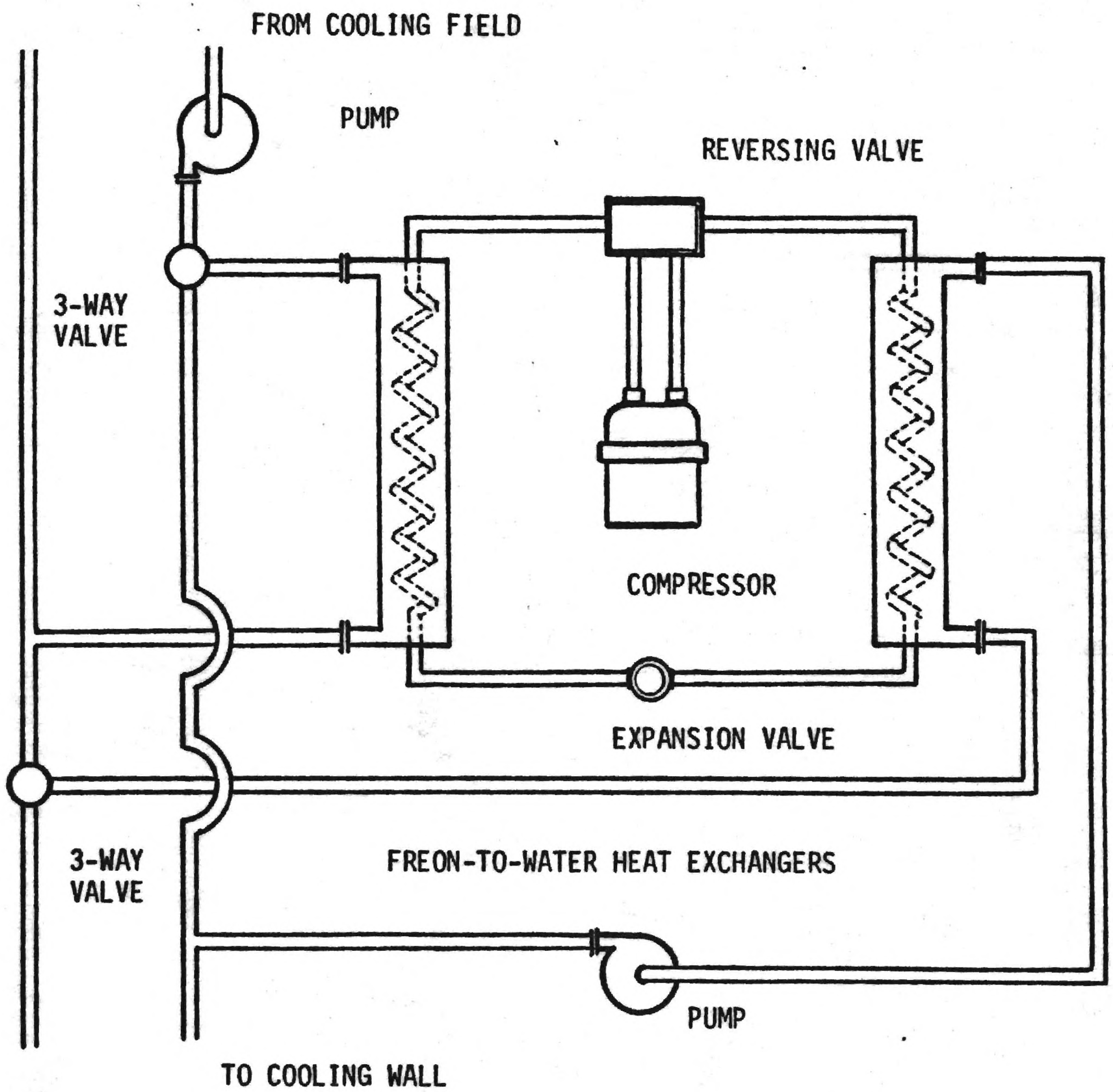


Figure 7 Passive/active cooling system

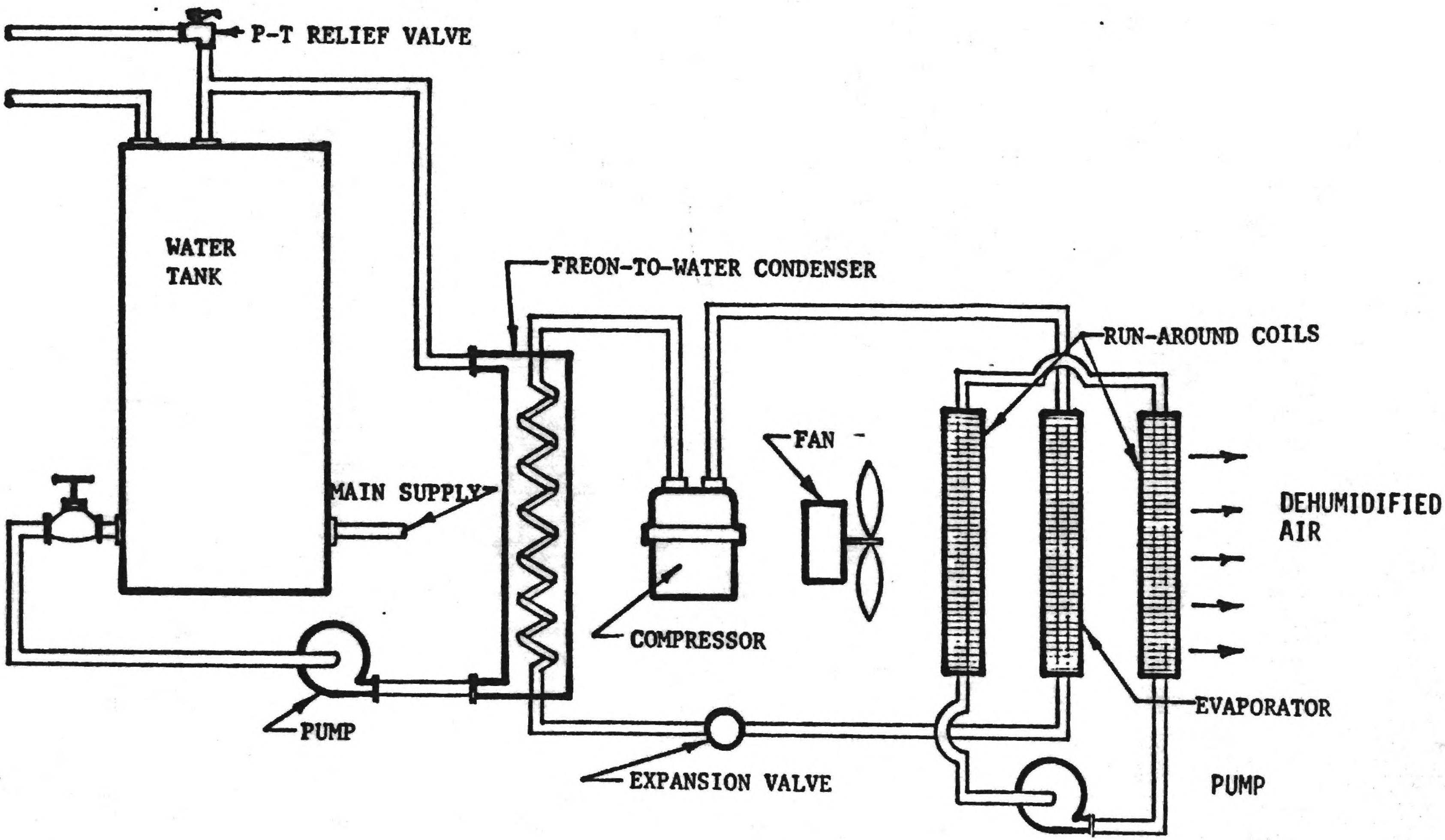


Figure 8 Modified heat pump DHW heater

6. SUMMARY

This program has established that earth temperatures at the depth of the cooling coil can be depressed significantly below the temperature of soil at a similar depth which has not been cooled or insulated. It has shown that seasonal storage of the cooling capacity is feasible and that the cooling capacity can be utilized through a radiative cooling scheme.

The program has shown that a passive/hybrid technique is feasible for hot-humid climates and that this technique is not only compatible with auxilliary cooling and heating systems, it has the potential of significantly improving their performance.

Work during the coming year should develop additional data on performance potential, operating modes and the practical feasibility of integrating the passive and auxilliary systems.

7. ACKNOWLEDGEMENTS

The work reported here has been funded through the Department of Energy, Contract DE-AC02-79CS30238

8. REFERENCES

1. "Panel Heating and Cooling Systems," **SYSTEMS HANDBOOK**, American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc., 1980.

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3. Wright, Scott, "T-Node Thermal Network Simulation Program," Thesis for Masters of Architecture, Georgia Institute of Technology, August 1981.

APPENDIX B

"Investigation of Passive Cooling For Hot-Humid Climates"

Passive and Hybrid Solar Energy Program Update: August 9-12, 1981

INVESTIGATION OF PASSIVE COOLING TECHNIQUES FOR HOT-HUMID CLIMATES

GEORGIA INSTITUTE OF TECHNOLOGY
COLLEGE OF ARCHITECTURE
ATLANTA, GEORGIA 30332

CONTRACT NO. DE-AC02-79CS30238

JAMES M. AKRDIGE

PERIOD OF CONTRACT: BEGINS 7/15/79 ENDS 9/30/82

OBJECTIVE

The objective of this program is to review existing and potential passive cooling techniques which might be applicable in hot-humid climates. Passive cooling techniques which show promise are to be ranked according to potential. One or more of the most promising techniques are to be selected for detailed theoretical study. After the theoretical study the most promising technique or techniques are to be built and evaluted experimentally. The ultimate objective of the program is to develop a passive cooling technique for hot-humid climates which is practical and economically feasible. A major goal of the program is to develop a passive cooling technique which will eliminate a significant portion of a building's cooling load.

BACKGROUND

The literature search revealed little, either in terms of architecture or devices, anywhere in the world which showed promise as a passive cooling method for hot-humid climates. The most prevalent and effective cooling strategy is not a cooling concept but a load minimization technique. If air is already hot and humid, it is imperative that radiant loads due directly or indirectly to the sun be minimized or eliminated. Shading devices are prevalent and extensive throughout the hot-humid regions of the world.

The study did show that earth tempering, i.e., underground construction, could significantly reduce cooling loads in hot-humid climates if air infiltration could be controlled. Despite this potential, few examples of earth tempering in hot-humid climates exist. Unlike most passive cooling and heating techniques for other climates, earth tempering in hot-humid climates has been held back by a technical problem. Until recently, it was not possible to control infiltration sufficiently well to make earth tempering practical in hot-humid climates.

The importance of infiltration or ventilation control in hot-humid climates become very apparent when one looks at the latent and sensible loads of a building as a function of the ventilation (infiltration) rates while keeping the ventilation air temperature constant. If the relative humidity of the air is now varied, one finds the only difference in the thermal load of buildings in arid and humid climates is due to the latent loads caused by ventilation (infiltration). Obviously, if infiltration can be greatly reduced and carefully controlled, a building in a humid climate will perform nearly the same as in an arid climate.

Although the study showed significant potential for earth tempering, it also showed serious architectural and market constraints on conventional earth tempered (underground) buildings. The study also showed that many of the thermal advantages of underground construction may be realized above ground through the use of a concept we have chosen to call "Detached Earth Tempering". If architectural constraints prevent taking the building underground, the Detached Earth Tempering Concept attempts to bring the thermal advantages of underground structures to above ground buildings.

The basic concept behind detached Earth Tempering is to embed coils in the earth through which water or other similar heat exchange fluids can pass. The fluid can then be circulated through building elements such as the floor, ceiling or walls. If the walls are well insulated and the insulation is located on the outside of the structure, one will have a cool wall structure similar to that of an underground structure. If infiltration is controlled through the use of good seals, vapor barriers and air locks at the doors and ventilation is accomplished through the use of an enthalpy exchanger, the building will perform similar to an underground structure in an arid climate.

Initial computer studies showed that ground temperatures at depths of 1.2--3.6 meters (4--12 feet) are much too high in late summer to provide significant cooling. The theoretical studies were verified with ground temperature measurements. The measurements verified the predicted ground temperatures for areas well shaded. They also showed that ground temperatures for areas with little ground cover can reach considerably higher temperatures. Figure 1 shows predicted ground temperatures in late August compared to temperatures measured in a shaded and unshaded area. The high temperatures result from propagation of ambient air and ground surface temperatures to the lower depths.

If high surface temperatures result in high temperatures at greater depths one might minimize the effect by separating the surface from the lower depth with insulation. Insulation of the soil from the surface also greatly reduces the rate at which energy can be lost to the ambient air during the winter months. This then requires that the soil beneath the insulation be cooled during the winter months if one is to have the low soil temperatures desired during the summer months. Georgia Tech has installed an experimental field with 213 meters (700 feet) of 38 mm (1.5 inch) polyethylene pipe buried at a depth of 1.2 meters (4 feet) with .9 meters (3 feet) of dirt directly above followed by 51 mm (2 inches) of extruded polystyrene insulation. The insulation is covered with

SOIL TEMPERATURE PROFILE

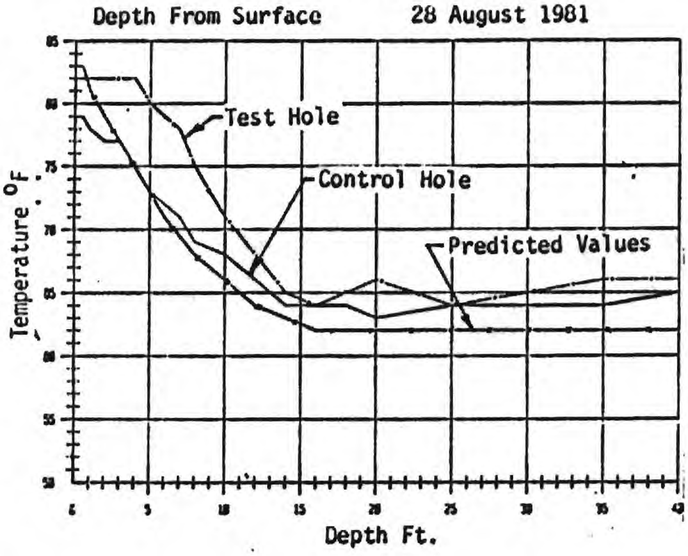

Figure 1. Soil Vertical Temperature Distribution

.3 meters (1 foot) of dirt wih a good sod cover. Ideally the field would be placed beneath the house to minimize undesirable ambient loads. Figure 2 shows a section of the Georgia Tech experimental field. Since it could not be installed beneath a house, great care has been exercised in providing a good sod cover to minimize radiant gains to the soil surface.

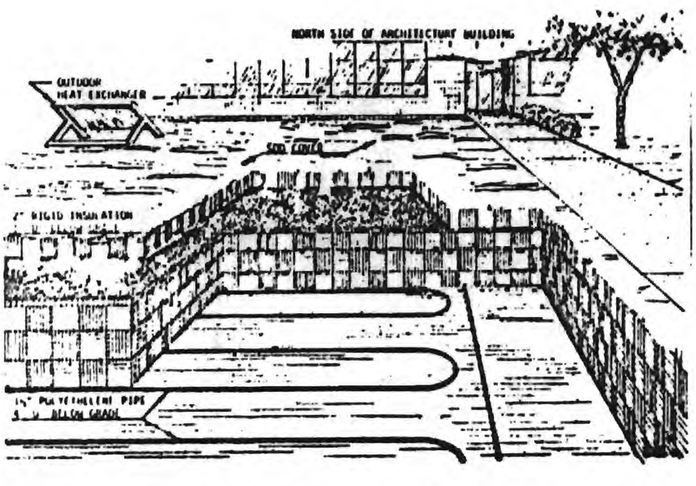

Figure 2. Crossection of Cooling Field

The insulated field is cooled during the winter months by circulating water through an above ground air-to-water heat exchanger and then through the buried coil. A differential thermostat turns on a small (1/15 hp) pump when the field is warmer than the ambient air. Data taken over the past winter indicate the air-to-water heat exchanger can operate without fan forced convection until the soil temperature approaches ambient temperature.

SUMMARY

The program has identified Detached Earth Tempering as a potentially promising concept for passive cooling in hot-humid regions. Computer similations have shown the concept to be capable of carrying a significant portion of the sensible cooling load. Experimental data now support the concept of cooling an insulated block of earth to well below the temperature of noninsulated soil at a similar depth.

The study has shown that buildings in hot-humid climates can be made to perform similar to buildings in hot-arid climates through the proper control of ventilation (infiltration). This greatly reduces the latent load normally associated with cooling in hot-humid climates. The study has also shown the feasibility of radiantly cooling above ground buildings using water cooled by an underground block of earth which was cooled the previous winter. Although a Detached Earth Tempered building does not exhibit all of the characteristics of underground buildings it does passively cool.

TECHNICAL ACCOMPLISHMENTS

The program has established that ground temperatures at the 1.2--3.6 meters depth (5--12 feet) are too high during late summer to provide significant cooling, either for earth cooling tubes or for the Detached Earth Tempering Concept.

The program has established that ground cover plays a significant role in establishing summer surface and subsurface temperatures. Good grass cover is better than bare earth while good tree cover is better than good grass cover.

The program has established that earth temperatures at the depth of the cooling coil can be depressed below the temperature of soil at similar depths which has not been cooled or insulated. Figure 3 shows a temperature profile across the cooling field showing how the temperatures on either side of the field are significantly higher than those within the field.

The program has established that during the charging cycle a natural convection air-to-liquid heat exchanger may be satisfactory until the ground temperatures begin to approach the air temperatures. This means that fans on the air-to-liquid exchanger need to only be run during the later part of the winter.

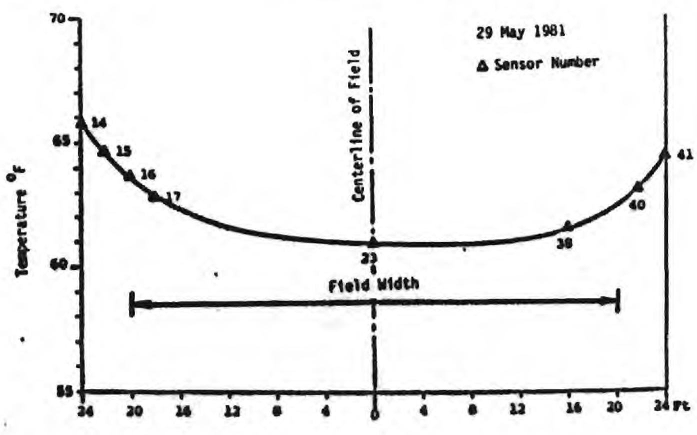

Figure 3. Temperature Distribution Across Field

The program has shown that the performance of the Detached Earth Tempering Concept could be improved if the earth block was not insulated during the winter and was insulated during the summer.

The program has shown that through proper control of ventilation (infiltration) buildings in hot-humid climates can be made to perform as though they were in an arid environment. This significantly affects the role of the auxilliary cooling (dehumidification) system.

The program has shown that through the use of building elements such as walls, roofs, or ceilings; buildings spaces may be made comfortable without cooling air temperatures nearly as low as necessary for cooled air system. This permits the use of cooling water at much higher temperatures than is possible with cooled air systems.

The program has demonstrated that coil placement in massive building elements such as concrete walls or floors may be critical if excessive temperature gradients are to be avoided. This is especially critical in buildings where dew-point temperatures are not held low. Improper coil placement in massive low conductivity elements used in buildings with high relative humidities may result in condensation directly opposite the coil without having sufficient cooling capacity in the building element.

The program has not identified any promising passive cooling techniques for handling latent loads which are not being actively pursued by other researchers. Latent loads may be more easily and efficiently handled by mechanical refrigeration equipment.

FUTURE ACTIVITIES

The cooling potential of the cooled earth block is being used to carry a thermal load programmed into an inline resistance heater. This part of the program will permit an experimental determination of the effectiveness of storing cooling potential in a block of earth insulated from the surface.

The program is also determining the effectiveness of different room radiative cooling schemes. This is initially determined in a 4 x 4 x 4 panel test box and will be expanded to a room size test chamber. These tests will permit optimization of coil location and spacing, optimization of radiator position and mass as well as determination of the radiative cooling potential of full size walls.

The success of many passive heating and cooling concepts is frequently compromised by very inefficient auxilliary heating or cooling systems. Work will be directed toward determination of the most efficient auxilliary cooling device and mode of operation. Initial calculations using a water-to-water heat pump with an elevated evaporator temperature (65^{o}F) indicates that a very high coefficient of performance is possible. High evaporator temperatures only become possible when the heat pump is used as a source of chilled water for a radiative cooling system.

A by-product of these tests will be additional insight into the convective transfer coefficients involved in room size walls which are heated or cooled.

Post-project Activities

Numerous builders, designers and architects have been in contact with these researchers asking for information which will permit them to include the concept in future designs. These people have been discouraged from embarking on such designs until firm design data are available. It is expected that once sufficient design data are available several local buildings will be constructed.

PUBLICATIONS/REPORTS

1. Ruberg, K., Akridge, J., Benton, C., and Houston, M., "Detached Earth Cooling with Radiant Interior Building Elements," to be presented at the ISES Meeting, September, 1981.

2. Akridge, J., "Passive Cooling for Hot-Humid Climates," to be given at the International Passive and Hybrid Cooling Conference, Miami, Florida, November, 1981.

3. Akridge, J., "A Decremented Average Ground Temperature Method for Estimating the Thermal Performance of Undeground Houses," paper to be presented at the International Passive and Hybrid Cooling Conference, Miami, Florida, November, 1981.

4. Akridge, J., "Investigations of Passive Cooling Techniques for Hot-Humid Climates," Progress Reports 1-8.

APPENDIX C

"Passive Cooling for Hot-Humid Climates"

Passive/Hybrid Cooling Conference
November 6-16, 1981

PASSIVE COOLING FOR HOT HUMID CLIMATES

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ABSTRACT

Earth Tempering is a term presently used for structures which are either buried or semi-buried. These structures typically use the ground as an insulating element and as a high capacitance barrier for the dynamic attenuation of heat flux.

In the proper environmental context, an earth tempered building can perform quite well. Unfortunately, underground buildings can be expected to meet considerable market resistance due to a number of factors. These include an increase in construction costs, waterproofing criteria, excessive site modifications and mass public resistance to subterranean housing. This has led to a possible cooling concept which the authors call "Detached Earth Tempering" (DET). The scheme involves an intimate but indirect coupling of selected building elements to the earth as a heat sink. Successfully applied, the concept promises the advantages of below-ground earth tempered construction without the structural, moisture, site and cost liabilities usually associated with below-grade buildings. This paper addresses the advantages and disadvantages of Detached Earth Tempering and describes a research program which is currently investigating the concept. The paper presents data on measured performance and predicted performance.

1. INTRODUCTION

Passive cooling in hot-humid climates has proven to be one of the most difficult challenges for natural building systems. Numerous literature searches have revealed little, either in terms of architecture or devices, anywhere in the world which shows promise as a passive cooling method for hot-humid climates. The most prevalent and effective strategy is not a cooling concept but a load minimization technique. If air is already hot and humid, it is imperative that radiant loads due to the sun be minimized or eliminated. Shading devices are prevalent and extensive throughout the hot-humid regions of the world.

Our study did show that earth tempering, in the form of underground construction, could significantly reduce sensible cooling loads in hot-humid climates if air infiltration could be controlled. Despite this potential, few examples of earth tempering in hot-humid climates exist. Unlike most passive techniques, earth tempering in hot-humid climates has been held back by a technical problem. Until recently, it was not possible to control infiltration sufficiently well to make earth tempering practical in hot-humid climates.

The importance of infiltration or ventilation control in hot-humid climates becomes very apparent when one looks at the latent and sensible loads of a building as a function of the ventilation (infiltration) rates while keeping the ventilation air temperature constant. If the relative humidity of the air is now varied, one finds the only difference in the thermal load on buildings in arid and humid climates is due to the latent loads caused by ventilation (infiltration). Obviously, if infiltration can be greatly reduced and carefully controlled, a building in a humid climate will perform nearly the same as in an arid climate.

Although the study showed heating and cooling potential for earth tempering, it also showed serious architectural and market constraints on conventional earth tempered (underground) buildings. The study also showed that many of the thermal advantages of underground construction may be realized above ground through the use of a concept we have chosen to call "Detached Earth Tempering" (DET). If architectural constraints prevent taking the building underground, the Detached Earth Tempering Concept attempts to bring the thermal advantages of underground structures to above grade buildings.

2. BASIC CONCEPT

The basic concept behind Detached Earth Tempering is to bury coils in the earth through which water or other similar heat transfer fluids can circulate. The fluid having been cooled by earth contact can then be circulated through building elements such as the floor, ceiling or walls. If the walls are well insulated and the insulation is located on the outside of the structure, one will have a cool wall structure similar to that of an underground structure. If infiltration is controlled through the use of good seals, vapor barriers and air locks at the doors and ventilation is accomplished through the use of an enthalpy exchanger, the building will perform similar to an underground structure in an arid climate.

Initial computer studies showed that ground temperatures at depths of 1.2--3.6 meters (4--12 feet) are much too high in late summer to provide appreciable cooling. The computer studies were checked with ground temperature measurements. These measurements verified the predicted ground temperatures for areas well shaded. They also showed that areas with little ground cover can reach considerably higher temperatures.

Because high surface temperatures result in high temperatures at greater depths, one might minimize this effect by separating the surface from the lower depths with insulation. Insulation of the soil from the surface also greatly reduces the rate at which energy can be lost to the ambient air during the winter months. This then requires that the soil beneath the insulation be cooled during the winter months if one is to have the low soil temperatures desired during the summer months.

Georgia Tech has installed an experimental field with 213 meters (700 feet) of 38 mm (1.5 inch) polyethylene pipe buried at a depth of 1.2 meters (4 feet) with .9 meters (3 feet) of dirt directly above, followed by 51 mm (2 inches) of extruded polystyrene insulation. The insulation is covered with .3 meters (1 foot) of dirt with a good sod cover. Ideally the field would be placed beneath the house to minimize undesirable ambient loads. Figure 1 shows a section of the Georgia Tech experimental field. Since it could not be installed beneath a house, great care has been exercised in providing a good sod cover to minimize radiant gains at the soil surface.

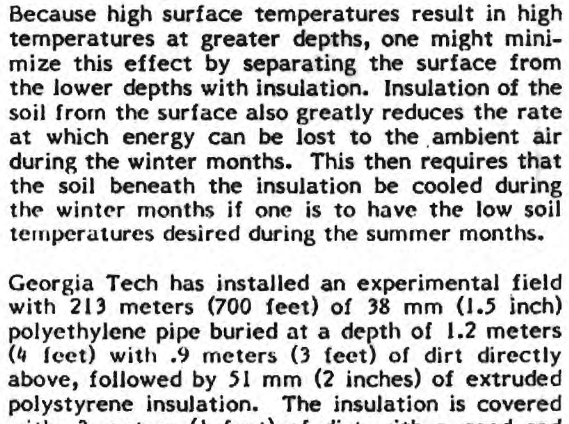

Fig. 1 Field cross-section

The insulated field is cooled during the winter months by circulating water through an above ground air-to-water heat exchanger and then through the buried coil. A differential thermostat turns on a small (1/15 hp) pump when the field is warmer than the ambient air.

The success of seasonal storage of cooling potential is highly dependent upon how one couples this cooling capacity to the occupants. Due to the low grade (temperature differences are relatively small) cooling potential, conventional cool air systems will not perform satisfactorily. Cooling through the use of building elements such as walls, floors or ceilings appears to offer the most potential. These elements provide large heat exchange surfaces in direct radiant contact with building occupants.

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Radiative cooling potential of concrete walls of several thicknesses with several different tube spacings have been simulated using a thermal simulation program called MITAS[2] and a smaller thermal network program for microcomputers called T-NODE[3]. These simulations show the radiative cooling concept to have potential. The simulations have also shown the need for experimental data on the performance of such walls due to uncertainties about convective heat transfer coefficients on cooled walls.

A radiative panel test chamber has been designed, constructed and is currently being used to develop experimental data on the performance of cooling walls. Figure 2 is an exploded view of the radiative cooling test chamber. This chamber has the capability of quantifying the cooling performance of walls, floor and ceiling elements.

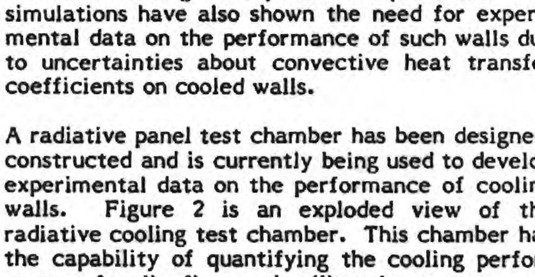

Fig. 2 Cooling panel test box

3. SYSTEM PERFORMANCE

Due to completion of the field installation and control system late last fall, combined with the high ground temperatures trapped beneath the insulation when the insulation was installed in late September, the field temperatures were not dropped to the levels last winter that they would reach during a normal cycle. Figure 3 shows temperatures across the field at a four foot depth for several dates. Notice on March 15th that the ground temperatures adjacent to the field (thermocouples number 14 and 41) are approximately the same as those within the field. This would not be expected when the field has run through a complete cycle.

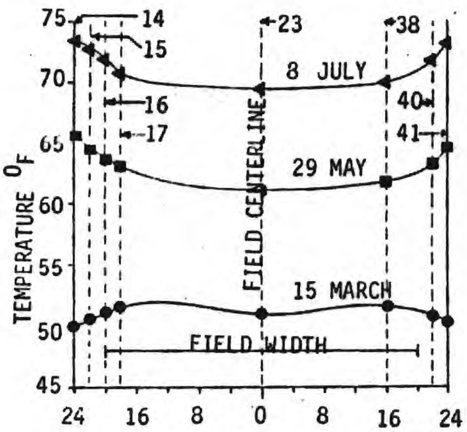

Fig. 3 Temperatures across the field

Figure 4 shows temperatures along a vertical shaft at the center of the field at the same time as the data given in Figure 3. Note that temperatures at depths below the four foot level increase with depth. Notice that the field temperature on May 29 in figure 4 is well below the adjacent ground temperature but is substantially above its temperature on March 15. At first this rapid rise in temperature was quite disturbing until one realized the tremendous sink below the field that was at a temperature well above the field temperature on March 15. This is shown quite clearly in Figure 4. Although some energy is diffusing from the surface through the insulation and warming the field, the field is also cooling the block of earth beneath it.

The field was scheduled to begin cooling a simulated building, using a resistance heater and load programmer, in early May. The load programmer failed during the initial checkout and couldn't be replaced until June 12 causing the load simulation tests to be delayed until that date.

The measured load for an existing energy efficient, but conventional design, home located in Columbus, Georgia was programmed in the simulator. The field has been carrying the entire

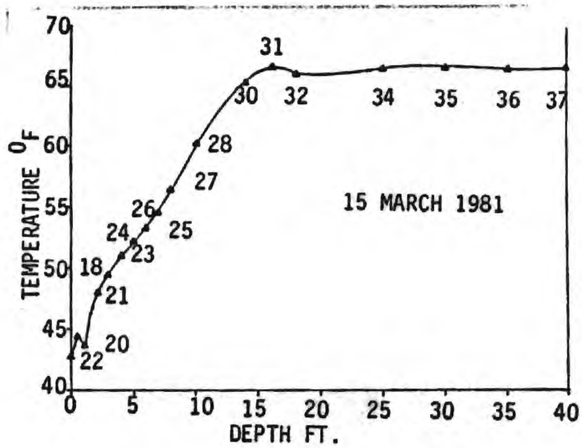

Fig. 4 Field temperature vs depth

sensible load of the building except for several days when the circulating pump lost prime and a leak in the field caused a brief shutdown. The July 8 curve on Figure 3 shows the field temperature is beginning to approach the point where it will be unable to provide water at a temperature low enough to radiatively cool. This early depletion of cooling potential was expected due to the late cooling start last winter and the high ground temperature resulting from the insulation being installed when the ground was the hottest.

4. AUXILLIARY SYSTEMS

It is imperative that passive cooling systems work well, or at least not interfere, with passive heating systems. It is also important that both passive heating and cooling systems be complemented with efficient auxilliary heating and cooling systems. Nothing useful is accomplished if much of the energy one saves with passive heating and/or cooling systems is lost through the use of inefficient auxilliary systems. Unfortunately many advocates of passive systems are opposed to incorporation of state-of-the art or high technology mechanical systems as a backup. This usually results in poor efficiency and less comfort.

It was felt from the start of this program that it would be highly unlikely that a passive cooling system for hot-humid climates could be developed that would be capable of meeting 100% of a residential cooling load. This meant that an auxilliary cooling system was necessary if comfort was to be maintained. It was also felt that passive techniques for meeting latent cooling loads are not likely to be developed in the near future. If the sensible cooling load of a building is to be met radiatively in a humid climate it is imperative that latent loads be efficiently handled. This means that auxilliary systems are needed to handle latent only at times and sensible and latent during extreme periods.

Unfortunately, if one were to employ a conventional air conditioner to handle the latent load, it would also provide sensible cooling which could be provided passively. It appears that greater efficiency can be obtained by handling the sensible and latent loads with separate equipment rather than with a single component as is normal practice.

4.1 Auxilliary Sensible Load

If one plots an idealized Rankine cycle air source air conditioner on a pressure enthalpy diagram one would have a cycle such as shown in Figure 5. The coefficient of performance (COP) would be about 3.42 for the most efficient systems presently available. One is limited to this COP by two factors. First, due to high ambient temperatures, one must have condenser temperatures of 150°F or above to dissipate the energy removed from the residence to the ambient air. One also must have evaporator temperatures at or below 50°F if one is to adequately handle the latent load.

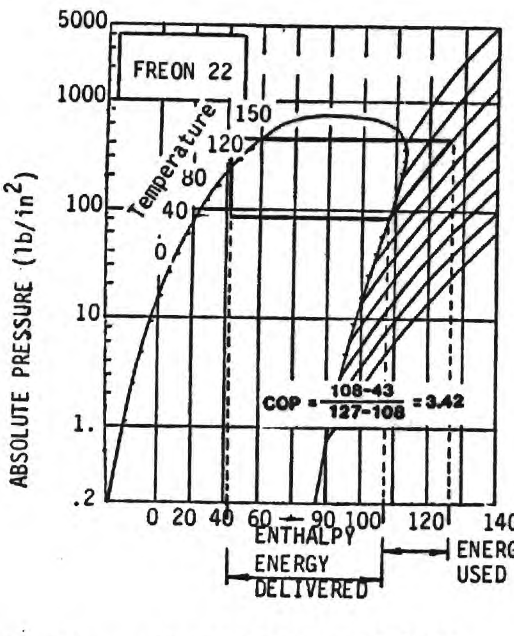

Fig. 5 Conventional heat pump-cooling

Use of a conventional air-source heat pump would not meet our stated desire to handle the sensible and latent loads separately. One can meet the sensible cooling load passively until the temperature of the water coming from the cooling field reaches 74-75°F. If one now supplies the 74-75°F water coming from the field to the condenser of a water-to-water heat pump and supplies water from the heat pump evaporator to the cooling walls, one can function with a Rankine cycle similar to the one shown in Figure 6. Notice it is not now necessary to operate the condenser at 150°F because of the 74-75°F water available from the field. It is also not necessary to operate the evaporator at 50°F because the radiative cooling wall works well with 70°F water. One now has a auxilliary sensible cooling system with a COP of 6.0. Figure 7 gives a schematic of the cooling field and cooling wall when operating through the water-to-water heat pump.

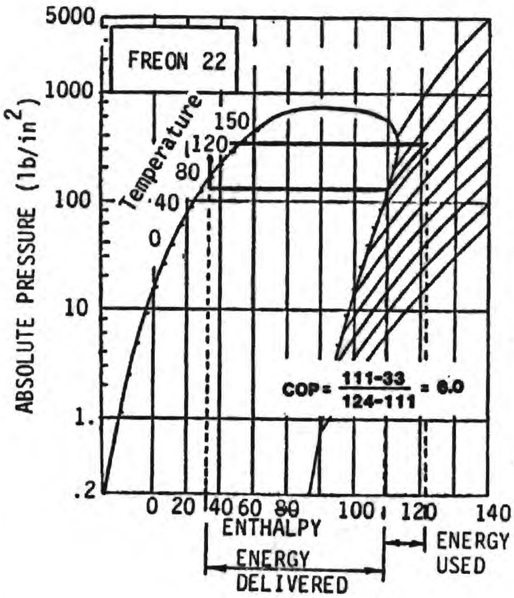

Fig. 6 Water-to-water heat pump

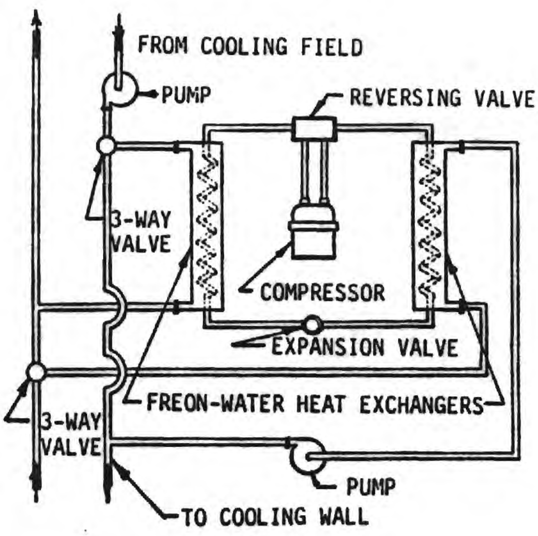

Fig. 7 System schematic

Similar cycles can be shown for a conventional heat pump and a water-to-water heat pump in the heating mode. One finds the COP improves from 3.10 to 6.92 by going to a water-to-water heat pump. This system obviously meets our requirement for a high efficiency sensible auxilliary system which is compatible with the passive system.

4.2 Auxilliary Latent Load

If one succeeds in passively heating and cooling a residence 100%, one finds that a substantial energy requirement still remains. It is not uncommon for domestic hot water (DHW) energy requirements to exceed the heating and cooling requirements for well designed energy efficient conventional homes. Several manufacturers have recently marketed domestic hot water heaters which operate on a heat pump principle. These heat pump DHW heaters require only 40-50% as much energy input as conventional electric resistance DHW heaters. If one locates the heat pump DHW heater in occupied space it not only heats the domestic hot water more efficiently, it also provides sensible and latent cooling. Figure 8 shows a heat pump DHW heater modified with a run-around coil. The run-around coil decreases the sensible cooling capacity and increases the latent cooling capacity without changing the total capacity or significantly affecting the efficiency of the heat pump as a DHW heater.

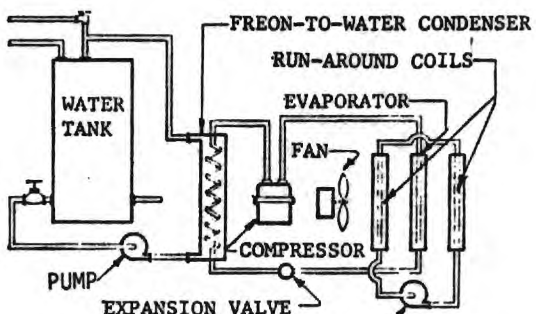

Fig. 8 Modified heat pump DHW heater

We now have an efficient DHW heater, a very efficient sensible auxilliary system and a latent auxilliary system which is a by-product of the DHW heater.

5. ADVANCE MODE OF OPERATION

Once the auxilliary heating and cooling systems have been integrated into the passive design, one finds that a second and possibly better mode of operation becomes possible. One can passively cool with cooling potential stored in a block of earth until the water from the cooling field reaches approximately 74°F. When the water reaches 74°F one actively cools with a water-to-water heat pump using the relatively cool 74°F water from the cooling field. This increases the field temperature until it reaches perhaps 110°F by the end of the summer. One can now passively heat using the 110°F water coming from the field and the radiative cooling/heating wall. When the water coming from the field reaches approximately 85°F, the water is directed through the water-to-water heat pump and the heat pump used to heat through the radiative wall. This cools the field until at the end of the winter the field has been cooled to perhaps 40-50°F. The system is now ready to begin another complete cycle. One now finds that the air-water heat exchanger described earlier and shown in figure 1 is not needed under the new operating mode.

Obviously the cycle will not operate exactly as described due to energy diffusion during the spring and fall. Energy diffusion only changes the temperatures given and not the validity of the proposed operating mode.

6. SUMMARY

This program has established that earth temperatures at the depth of the cooling coil can be depressed significantly below the temperature of soil at a similar depth which has not been cooled or insulated. It has shown that seasonal storage of the cooling capacity is feasible and that the cooling capacity can be utilized through a radiative cooling scheme.

The program has shown that a passive/hybrid technique is feasible for hot-humid climates and that this technique is not only compatible with auxilliary cooling and heating systems, it has the potential of significantly improving their performance.

Work during the coming year should develop additional data on performance potential, operating modes and the practical feasibility of integrating the passive and auxilliary systems.

7. ACKNOWLEDGEMENTS

The work reported here has been funded through the Department of Energy, Contract DE-AC02-79CS30238

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APPENDIX D

"A Decremented Average Ground Temperature
Method for Estimating the
Thermal Performance of Underground Houses"

Passive/Hybrid Cooling Conference
November 6-16, 1981

A DECREMENTED AVERAGE GROUND TEMPERATURE METHOD FOR ESTIMATING THE THERMAL PERFORMANCE OF UNDERGROUND HOUSES

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ABSTRACT

Underground or earth tempered houses may be one of the more satisfactory approaches to passive cooling for much of the United States. They have already demonstrated good performance and significantly reduced thermal loads in applications where heating is the primary load. Since ground temperatures are much more moderate than air temperatures in all parts of the country, peak building loads will be less for underground buildings.

While underground or earth tempered houses are being built in ever increasing numbers, few have been analyzed to predict thermal performance. Large computer simulation programs are required for rigorous analysis, resulting in prohibitively large costs.

While the Average Ground Temperature Method, the Underground Degree Day Method, and the Ground as an Insulator Method have provided very rough estimates of thermal performance, all have ignored the effect the building has on the ground temperatures. This leads to underestimating the thermal performance during the heating season and overestimating the performance during the cooling season.

This paper presents a method which, although still an estimate, takes into account the effect of the building on the ground temperatures. The method allows one to determine the effect of insulation, depth, soil properties and time on the performance of the building.

1. INTRODUCTION

Earth sheltering of structures is a particularly appropriate way to design in climates where extreme temperatures are common because the earth responds very slowly to changes in ambient air temperature. Due to the time lag caused by the mass of the earth around a building, an above grade heating peak will occur approximately six weeks before the corresponding peak is reflected at six feet below grade. The same time lag that helps to reduce winter heating peaks is useful for cooling in the summer, and if properly used can minimize the need for mechanical air conditioning. Soil temperature does fluctuate seasonally, and the total yearly temperature change decreases with greater depth, until at a depth of approximately thirty feet the temperature remains constant throughout the year.

The only accurate method for predicting the thermal performance of underground structures is through the use of large computers and finite difference equations. Unfortunately these are not available to everyone, so estimation methods must be used. Three methods for estimating thermal loss from underground surface have been reported in the literature. These are: Average Ground Temperature Method(1), Subsurface Degree Day Method(2), and Earth as an Insulator Method(3). All three of these methods ignore the effect of the Building on the adjacent soil temperature. This results in these methods overestimating heating loads and underestimating cooling loads. This paper presents a method, called the **Decremented Average Ground Temperature Method,** which does account for the effect of the building on the adjacent soil temperature.

A brief review of the methods commonly used will point out their advantages and disadvantages and the need for another approach to determining thermal loads for underground structures.

1.1 Average Ground Temperature Method

This method, used by the state of Minnesota for comparing the performance of underground buildings with the performance of other passive and low energy houses is based on the assumption that energy loss from underground buildings can be calculated much as one would an above ground building. Equation (1) assumes that energy loss is proportional to the undisturbed or far field ground temperature.

$$Q = (T_r - T_e) * U * A \tag{1}$$

This method assumes that the building will not affect the undisturbed ground temperature, i.e., the ground will conduct the energy away from the building as fast as the building loses energy. As we will see later, this is an erroneous assumption but the method does lay the groundwork for a more accurate method. The method is presented here so the reader will be familiar with the method, because it is very easy to use and because it will give conservative heating loads, i.e., the building will perform better than predicted. Unfortunately the method gives optimistic cooling load predictions, i.e., the earth will not provide the cooling predicted.

The undisturbed or far field ground temperatures used in equation (1) must be calculated for the particular location where the structure is located. Fluker[4], Moreland[5], Ingersol[6] and Labs[7] have presented equations for predicting undisturbed ground temperatures as a function of Location, time and soil properties. Table I gives soil temperatures at various depths for Atlanta. This table was developed using the equation listed at the bottom of the table. The equation developed by Ken Labs has proven to be easy to use and quite accurate, provided the ground is covered with grass or is well shaded. Soils exposed directly to the sun, experience much higher summer temperatures.

1.2 Subsurface Degree Dry Method

The subsurface degree day method developed by Ken Labs [7] arrives at an estimate of monthly or annual heating or cooling load based on the reduction of annual degree days using the average soil temperatures. The method is primarily useful in estimating quantitatively the effect of subsurface constuction on the annual heating and cooling load. As with the previous method, the SSDD method does not take into effect the building influence on the undisturbed soil temperatures. This method, as with the previous one, results in a conservative estimate of the heating thermal load, and underestimates the cooling thermal load. The SSDD method is essentially the Average Ground Temperature Method converted to a degree day loss. Both methods give the same results over a monthly or seasonal time period.

1.3 Earth as Insulation Method

While the previous two methods calculate thermal loads on underground surfaces based on building surface thermal properties and undisturbed soil temperatures, the Earth as an Insulation Method use average monthly or seasonal ambient temperatures and calculates an equivalent thermal resistance based on the building and ground thermal properties and surface depth and orientation.

While the average profile temperature provides a rough indication of the severity of ground climate, most heat transferred from shallow underground walls will occur between the wall and the surface, rather than horizontally to the subsoil. ASHRAE's 1977 **Fundamentals Handbook** suggests a procedure for estimating heat loss from subgrade walls via a radial heat flow path; it assumes a steady-state exchange of heat from the wall to a constant monthly or seasonal ambient temperature.

Blick[8] has shown that this method works quite well for roofs where the energy flow path is well understood. For a roof surface the thermal loss or gain may be expressed as:

$$Q = \frac{\bar{T}_o - T_r}{R_e} \qquad (2)$$

It must be emphasized that equation (2) is based on the assumption that hourly temperature profiles are damped out by the soil above the roof, i.e., the roof only sees the average temperature. This assumption is reasonably valid for soil thickness greater than 1½ - 2 feet.

For vertical sufaces such as a wall it is assumed that the energy flows in an arc with the center of the arc located at the ground surface and projected wall surface intersection. Labs[7] gives equation (3) to calculate average thermal resistance of

TABLE I GROUND TEMPERATURES IN UNDISTURBED SOIL

ATLANTA, GEORGIA

DEPTH (FT)	MONTH JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
SURFACE	44	43	48	56	66	76	82	83	78	70	59	50
2	50	47	49	54	62	70	76	79	77	72	64	56
4	55	51	51	54	59	65	71	75	75	72	67	60
6	58	55	53	54	58	63	67	71	73	71	68	63
8	61	58	56	56	58	61	65	68	70	70	68	65
10	63	60	58	57	58	60	63	66	68	69	68	66
12	64	62	60	59	59	60	62	64	66	67	67	66
14	65	63	61	60	59	60	61	63	65	66	67	66
16	65	64	62	61	60	60	61	62	64	65	66	66
18	65	64	63	62	61	61	61	62	63	64	65	65
20	65	64	63	62	62	61	61	62	63	64	64	65

(1) These temperatures were calculated for sod covered ground using the following equation and constants:

$$T_{x,t} = Tm - (As\ e^{-x(\pi/365\alpha)^{\frac{1}{2}}}) * \cos \frac{360}{365} (T - To - \frac{x}{2} (\frac{365}{\pi\alpha})^{\frac{1}{2}})$$

subsurface walls.

$$\overline{R}_{e(x-y)} = \pi R_{soil} * \left(\frac{x^2 - y^2}{x-y}\right) * \left(\frac{90 - m}{360}\right) + R_w \qquad (3)$$

Substitution of Re(x-y) for Re in equation (2) allows one to calculate thermal loses or gains from underground walls.

As with the two previous methods, this method ignores the effect of the building on the adjacent soil temperature. It also ignores thermal lag due to the soil since the loss is calculated based on the average ambient temperature for the particular time evaluated.

2. DECREMENTED AVERAGE GROUND TEMPERATURE METHOD

The **Average Ground Temperature Method** discussed earlier is easy to understand and easy to use. Its primary disadvantage lies in its inability to consider the effects the building has on the undisturbed ground temperatures. The **Decremented Average Ground Temperature Method** (DAGT) described here modifies the undisturbed ground temperatures based on energy loss rate from the building and the ground thermal properties. The method does take into consideration effects which were ignored in the previous three methods, thus, should be more accurate.

Intersoll(6) showed that energy loss from an underground source could be defined by:

$$\left(T_{ts} - T_e\right) = \frac{Q'_{\lambda} r^{(2-n)}}{2\pi^{(n/2)} k} \int_{r\eta}^{\infty} \beta^{(n-3)} e^{-\beta^2} \, d\beta \qquad (4)$$

For linear flows such as from a wall where n=1 equation (4) reduces to:

$$\left(T_{ts} - T_e\right) = \frac{Q' r}{2K\sqrt{\pi}} \int_{r\eta}^{\infty} \beta^{-2} e^{-\beta^2} \, d\beta \qquad (5)$$

$$\left(T_{ts} - T_e\right) = \frac{Q' r}{2K\sqrt{\pi}} \left(\frac{e^{-r^2\eta}}{r\eta} - 2 \int_{r\eta}^{\infty} e^{-\beta^2} \, d\beta\right) \qquad (6)$$

If one now sets r=O, one obtains:

$$\left(T_{ts} - T_e\right) = \frac{Q'\sqrt{\alpha t}}{K\sqrt{\pi}} \qquad (7)$$

Equation (7) assumes that energy is being transferred from both sides of the plane. For a wall of an underground structure, energy is only lost to the ground from one side of the wall, so equation (7) becomes:

$$\left(T_{ts} - T_e\right) = 2\left(\frac{Q'\sqrt{\alpha t}}{K\sqrt{\pi}}\right) \qquad (8)$$

If we define the thermal resistance of the wall, including film coefficient and insulation, as R_w; the total resistance to energy loss becomes:

$$R_t = R_w + R_g \qquad (9)$$

$$R_t = R_w + 2\,\frac{(\alpha t/\pi)^{1/2}}{K} \qquad (10)$$

Energy loss from the wall now becomes:

$$Q_t = \frac{1}{\left(R_w + \frac{2\left(\frac{\alpha t}{\pi}\right)^{1/2}}{K}\right)} * A * \left(T_r - T_e\right) \qquad (11)$$

If we now set time t = O we can calculate the energy loss of the wall at time = 0, i.e., before the wall has begun to influence the soil temperature. Equation (10) at time = 0 reduces to:

$$Q_{t=0} = \frac{1}{R_w} * A * \left(T_r - T_e\right) = U_w * A * \left(T_r - T_e\right) \qquad (12)$$

Note! This is the same equation that was given for the Undisturbed Ground Temperature Difference Method. If we now divide equation (10) by equation (11), we obtain a decrement factor, f_d, which shows how the rate of energy loss from an underground wall decreases with time. We obtain:

$$f_d = \frac{R_w}{\left(R_w + \left(\frac{2\left(\frac{\alpha t}{\pi}\right)^{1/2}}{K}\right)\right)} \qquad (13)$$

The wall loss equation now becomes:

$$Q_t = f_d * \frac{1}{R_w} * A * \Delta T = f_d * U_w * A * \Delta T \qquad (14)$$

While it is possible to substitute equation (12) into equation (13) and solve for each case, it is much easier and educational to investigate how f_d varies as wall thermal resistance (R_w), soil conductivity (K_s), soil diffusivity (α) and time (t) vary. Figure 1 shows how f_d varies with time for given soil properties as a function of various wall thermal resistance. This figure shows that f_d drops much more rapdily for low wall thermal resistances, i.e., the building is disturbing the adjacent soil temperatures much more at low wall thermal resistances. While f_d is lower for lower wall thermal resistances one must not jump to the conclusion that energy loss is less for walls with low thermal resistances. Figure 2 shows how Q varies with time for various wall thermal resistances. Note that Q is always lower for higher wall thermal resistances but the magnitude becomes less at longer times.

Figure 3 shows the effect of soil thermal conductivity on f_d as a function of time while figure 4 shows the effect of thermal diffusivity on f_d as a function of time.

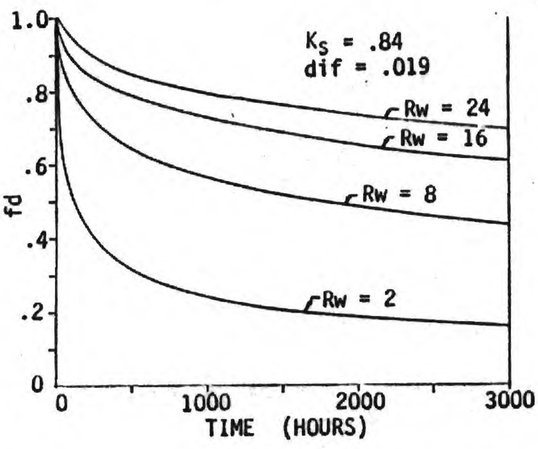

Fig. 1 Decrement factor as a function of time and wall thermal resistance

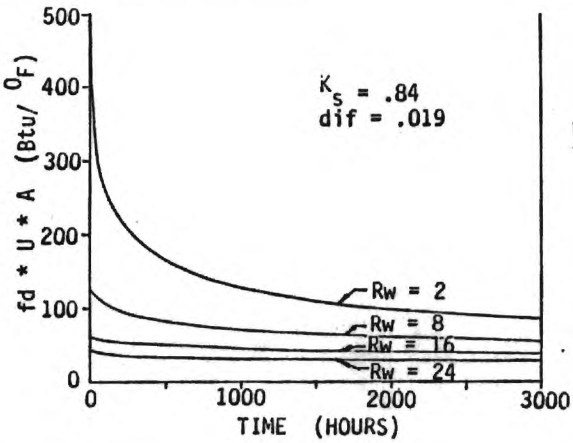

Fig. 2 Effect of time and wall thermal resistance on fd*U*A

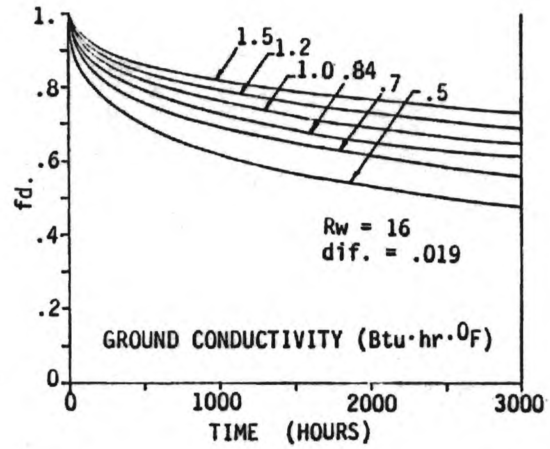

Fig. 3 Decrement factor as a function of time and ground thermal diffusivity

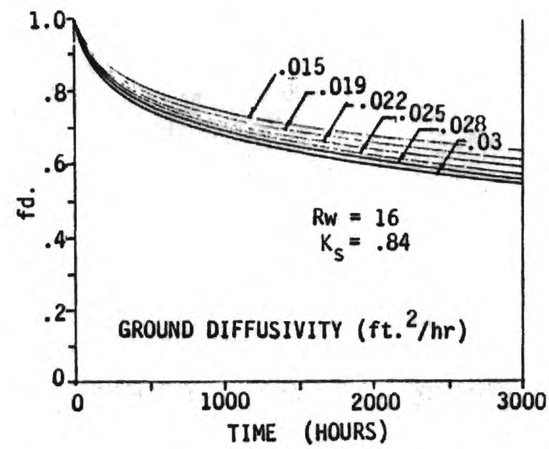

Fig. 4 Decrement factor as a function of time and ground thermal diffusivity

2.1 Limitation of DAGT Method

Although the Decremented Average Ground Temperature method does consider the effect of the building on adjacent soil temperatures and provides more accurate values than the other three methods, it presently has several limitations which should be addressed. As presently configured the DAGT method does not lend itself well to calculating the losses through earth covered roofs or bermed walls. A basic assumption of equations (8) through (9) is that Q remained constant. Later calculations show Q to decrease as a function of time and wall thermal resistance. Since equations 10 - 13 do not include the effect of the decreasing Q some error is introduced. This limitation results in f_d decreasing slightly more rapidly than in practice. f_d also continues to decrease at large values of time rathern than approaching an asymptote. No simple solution which will include the effect of the decreasing Q has been found, although numerous step function numerical solutions are possible. It is perhaps better to limit the maximum value of time to some value such as 2000 hours until additional verification can be made

3. COMPARISON OF METHODS

Figure 5 shows the loss from a 8 foot high wall section located an average of 8 feet below the surface as predicted by the four methods as a function of time. Figure 5 also shows the loss calculated by a computer program. Notice that the Average Ground Temperature Methods predicts the highest loss and remains constant. The Ground as an insulator method predicts a lower loss (also constant) than any of the methods until 200 hours. The loss predicted by the DAGT method and the computer program agree with the Average Ground Temperature Method at time zero with both dropping quite rapidly during the first 750 hours. At 3000 hours the Average Ground Temperature predicts a 5224 Btu/hr load, the Earth as an insulator method predicts a 1572 Btu/hr load, and the DAGT method predicts a 610 Btu/hr load. The subsurface Degree Day Method is not shown because it agrees exactly with the Average Ground Temperature method.

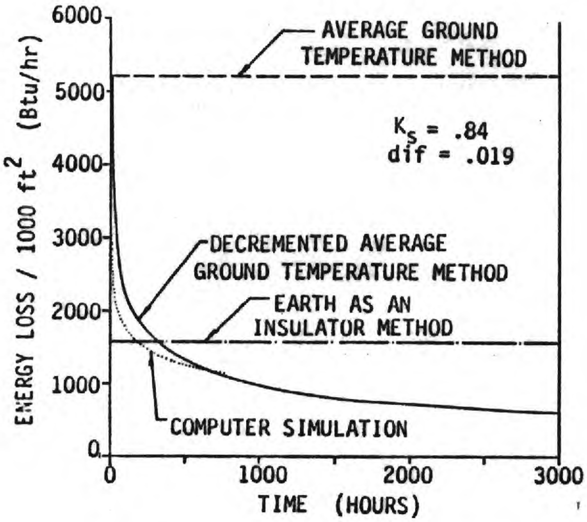

Fig. 5 Comparison of methods for estimating loss from subsurface walls

The Decremented Average Ground Temperature Method is obviously not a final solution. It is presented here to show that more accurate estimation methods are possible. It is expected that further work will result in a manual solution which includes the effect of the decreasing Q. Until a modified equation can be developed it is perhaps wise to limit time to 1500 to 2000 hours. The Earth As An Insulator Method appears to provide some improvement over the other estimation methods.

4. NOMENCLATURE

A	=	Surface area (FT^2)
As	=	Annual Surface Temperature Amplitude in ^oF
k	=	Thermal conductivity of soil Btu/(hr·^oF·FT)
m	=	Surface slope (degrees)
n	=	1 for a surface
Q	=	Thermal loss or gain (Btu/hr)
Q'	=	Rate of heat flow per FT^2 of surface (Btu/hr·FT^2)
Re	=	Equivalent thermal resistance = Rw + x_*Rs
$\overline{Re}$(x-y)	=	Average thermal resistance of a subsurface wall (hr·FT^{2}·^oF·/Btu)
Rs	=	Thermal resistance of soil per FT (hr·FT^oF/Btu)
Rw	=	Thermal resistance of building element
r	=	Tube radius (ft) = O for a surface
T	=	Day of year (Jan. 1 = day 1)
Te	=	Undisturbed or far field ground temperature at a given depth (^oF)
Tf	=	Phase constant, number of days after day 1 when minimum temp. occurs
Tm	=	Mean Annual ground temperature (^oF)
T_o	=	Average monthly ambient temperature (^oF)
Tts	=	Outside surface temperature (^oF)
T_r	=	Room temperature
t	=	Time after start (hrs.)
ΔT	=	(Tr - Te)
U	=	Thermal conductance (Btu/hr·FT^{2}·^oF)
U_w	=	Wall thermal conductance (Btu/hr·$FT^2$$^oF$)
X	=	Soil depth to top of wall or to a given depth (FT)
Y	=	Soil depth to bottom of wall (ft)
α	=	Thermal diffusivity of soil (FT^2/hr) or (FT^2/day)
β	=	Variable of integration
Cp	=	Specific heat of soil (Btu/lb·^oF)
ρ	-	Density of soil (lb/FT^3)
η	=	$\frac{1}{2\sqrt{\alpha t}}$

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APPENDIX E

"Detached Earth Cooling with Radiant Interior Building Elements"

DETACHED EARTH COOLING WITH RADIANT INTERIOR BUILDING ELEMENTS

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Atlanta, Georgia

ABSTRACT

Earth coupled cooling requires a heat exchange element in the building that maintains its cooling efficacy at relatively high (22.7°C) ground temperatures. Decoupling the earth from the building element and transferring heat from these interior surfaces to a cooling field with water, allows the occupant control over the temperature of the space and improves the cooling potential of the earth. Radiant wall panel heat exchangers are identified as viable options. Preliminary experiments with a concrete slab cooling panel indicates that mass incorporated in a radiant element yields uniform panel temperatures and maximizes cooling field potential. Radiant coupling to a secondary mass increases the response time of the slab and maintains cool temperatures in the secondary mass.

1. INTRODUCTION

Passively cooling a building by using the ground as a heat sink has been popularly accepted in the form of underground, or "earth-coupled" structures. Directly coupling building elements such as the walls, floor, or ceiling to the earth has numerous disadvantages, among them the lack of control over the temperature and heat transfer rates of the structural building elements. This lack of control leads to problems in hot-humid climates including condensation on walls in early spring, high wall temperatures in the fall and excessively cold surfaces in late winter. By decoupling the heat sink from the building element, control may be achieved over the building's interior temperature and the building element's heat flux to the earth.

Earth coupled radiative cooling comprises two elements: 1. an earth-coupled element and 2. a building side heat-exchanger. Polyethylene pipe buried four feet below the surface transfers heat from the building's interior to the ground. The block of earth associated with the pipe is separated from ambient temperatures by a two inch polystyrene layer at the one foot below grade. Temperatures in test field have ranged from 20°C in early spring to 22.7°C in late July with simulated building cooling loads applied to the field. Field temperatures limit the type of heat exchangers practicable in residences and curtail the amount of heat that may be instantaneously removed from a building.

Earth coupled cooling is characterized by low temperature differences between the earth and the space that requires cooling. For example, in Atlanta earth temperatures at the end of August at a depth of 2.44m are a. 20.5°C. This temperature when directly coupled to building walls will increase due to the transfer of heat from the building to the earth. These relatively high cooling temperatures preclude the use of convective elements for maintaining comfortable air temperatures. Unless coupled to a mechanical cooling mechanism capable of producing a large temperature potential between the interior air and heat exchange fluid, convective heat exchanges, when coupled with earth heat sinks, play a secondary role to radiant heat exchange.

Radiant space conditioning panels are capable of maintaining comfort conditions with higher temperatures than forced air systems due to the change in mean radiant temperature induced by the panels[1]. In an extreme example, comfort may be maintained by a mean radiant temperature of 22.7°C with a concurrent 33.9°C air temperature,in a room where a person is sitting (1 MET) wearing light clothes (0.9 CLO) while the surrounding air is moving at $0.1 ms^{-1}$ and relative humidity is 20%. The example illustrates the effectiveness of space conditioning with radiant surfaces.

The above example is overly simple and the following issues must be considered before panel design is tested:

1. Mean radiant temperature is not the temperature of a radiant heat exchanger, but is rather the weighted sum of all surface temperatures in a room. The placement and distribution of radiant panels must maximize radiant exposure of the panel.
2. Surface temperatures of the panel can only be maintained at low temperatures with respect to air temperatures if they have an extremely high heat transfer capability. If dT=10°C, between panel surface and air, then the heat transfer fluid must transfer a. $7.4 Wm^{-2}$ from the panel to the cooling source. A second

alternative to instantaneous heat transfer from the room to the fluid is to provide the panel with heat storage capacity.

Radiant heating systems were extensively used during the 1950's in residences. Copper pipe was typically laid in concrete slab floors and conditioned water kept the surface at a constant temperature. Noted for their comfort,[4] widespread application of the systems did not occur due to the relatively high installation costs when compared with forced air systems. When used in 0.15m thickness floor slabs, response time to instantaneous loads were reported as a major drawback of the system. Typical cooling applications in commercial buildings used light weight steel panels in ceilings to reduce the response time of the mechanical cooling system[2]. Water in these panels was mechanically cooled and because of the light weight and high conductance of the panels, the surfaces responded quickly to the peak cooling demands of commercial environments.

Although most radiant heat transfer surfaces have been located in floors and ceilings, both the floor and ceiling of a room present a smaller radiant surface than do walls to the sitting and standing postures of occupants. Although radiant cooling is not strictly similiar to radiant heating by room elements, analogies may be drawn between the two. For example, in a 4.9m x 4.9m room, under a 2.44m high radiant ceiling, a maximum of 2.5% of the heat is transferred from the ceiling to the occupant in a radiant heating system. A floor would radiate only 2.25% of its energy to the subject[3]. A person standing in the middle of the room will receive 6% of the radiant energy from two opposing walls, or 9% of the energy if all four walls are effective radiant surfaces. In addition to the comfort considerations, walls are effective

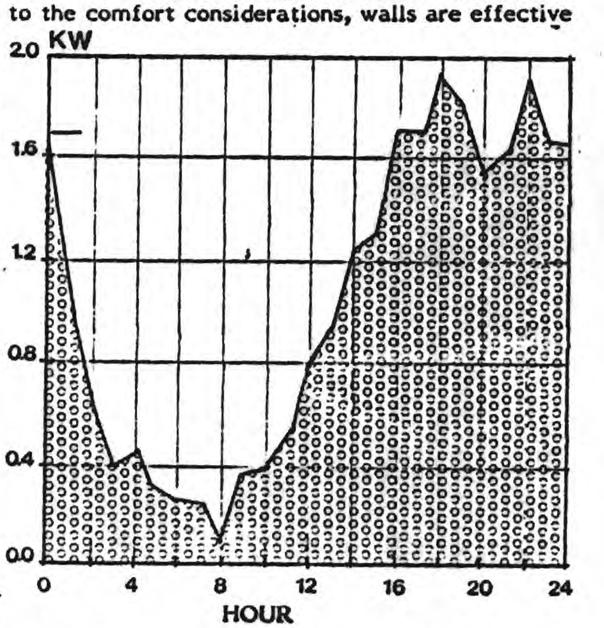

Fig. 1. Diurnal cooling loads in June for typical house in Georgia. These profiles are used to generate the diurnal load on the radiant cooling slab.

convective heat exchange surfaces due to the vertical attitude. Walls typically comprise the largest portion of the mean radiant surrounds and are the most effective building elements for interior heat exchange.

With tightly built, well insulated homes, both internal gains and the ambient climate combine to impose a cooling load on the air-conditioning system. A daily profile of the cooling load in a 158 m^2 house with a 3808 W design heat loss (with a temperature differential of 11.1°C) is illustrated in fig.1. With this type of daily cooling load profile, the instantaneous loads on a radiant cooling panel coupled to an earth-coupled cooling source would overwhelm the system's cooling capacity. For example, radiant cooling walls deployed in a typical dwelling would have to remove up to 2636W to the cooling field in order to maintain comfort.

Simulations indicate that the earth's capacity would be quickly depleted with this type of instantaneous load and the ability to meet cooling demands substantially reduced. The cooling load must be more evenly spread over the diurnal cycle in order to increase the field's effectiveness.

To maintain comfort conditions with the ground's high and slowly changing temperature source, a concrete radiant panel system was designed and tested for application in residences as the interior heat exchange element for the earth-coupled cooling field.

2. RADIANT PANEL PERFORMANCE

Successful application of a radiant wall cooling system requires matching the available earth cooling potential with the panel's heat transfer characteristics.

Criteria for radiant concrete panels include heat exchange fluid (water) flow rate, slab thickness, pipe spacing within the wall, and coupling to other masses in the space. Application of high-mass radiant panels in a residence is not common and will require careful detailing before application. Poured concrete walls with embedded copper pipe are non-standard residential construction elements, but have been used for testing purposes in this *study*. The concrete slabs have 0.009m diameter copper *pipe* embedded on the rear of the slab.

Detailing of a practicable residential radiant wall cooling system is seen as a future task once the proper performance specifications have been developed. For this pupose, a calorimetric box was constructed in the research space of the College of Architecture to measure the thermal performance of high-mass radiant panels.

Presently, only preliminary performance data have been recorded and the interpretation of the results of experiments is qualitative rather than rigorous. This initial battery of tests has revealed the usual debugging problems in an experimental process and the instrumentation, but has also yielded initial performance data of a 0.05 m concrete slab under both steady state and intermittent heat loads.

In order to isolate the thermal performance of the slab and provide an adaptable interior space for testing various wall combinations, the calorimetric box was constructed with 0.1 m moveable polyurethane walls. These walls define a constant 2.21 m^3 volume (1.22 m x 1.22 m x 1.22 m) and allow the addition of brick or drywall materials to the interior cube surfaces. Edge losses are minimized by 0.25m x 0.25m polyurethane members defining the edges of the cube. Conductance of the cube has been calculated at 1.46 W^oC^{-1}. Baffled resistance lighting in the cube imposes a cooling load on the radiant panel being tested.

The concrete slab used in the calorimetric box is a 1.22 m x 1.22 m x 0.05m slab with copper pipes spaced at .25 m (10 in) intervals on the panel's back. Water is circulated through the pipe at $4.88 \times 10^{-5} m^3 s^{-1}$ per square meter of radiant panel area. The thickness of the slab being tested represents only half a typical slab, with both sides exposed. Losses from the rear panel of the slab are negligible. Thermocouples are distributed in the calorimetric box and slab as shown in the exploded view of the calorimetric box in fig.2. Heat flux sensors have also been applied to the front face of the slab.

2.1 Performance of Radiant Walls

During this initial series of experiments, three attributes of high-mass radiant wall systems were observed: 1. the surface temperatures of the slab are not influenced by a change in flow rates for pipe flow rates tested; 2. the surface temperature of the slab is influenced by radiant coupling to secondary mass in the space; and 3. surface

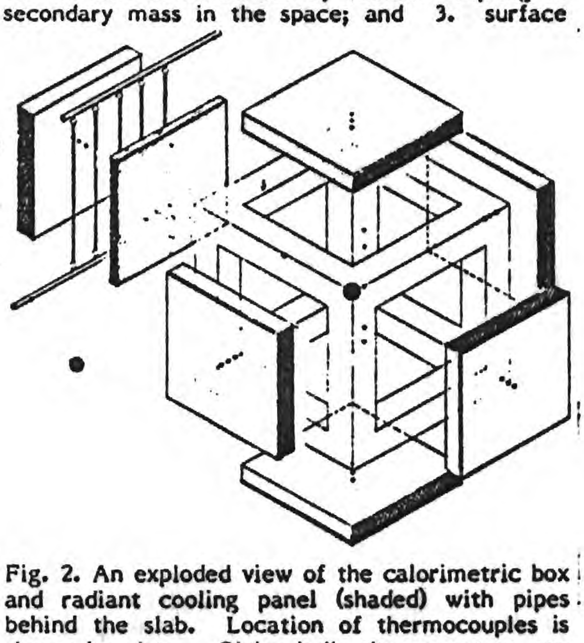

Fig. 2. An exploded view of the calorimetric box and radiant cooling panel (shaded) with pipes behind the slab. Location of thermocouples is shown by dots. Globe bulb thermometers are shown as shaded circles.

temperature distribution is uniform with diurnal cooling load profiles. This last attribute is crucial for maintaining comfort and heat transfer capability with wide pipe spacing and acceptable application costs. The slab was tested with its front facing the interior of the cube and its hardboard surfaces. A second test incorporated a solid core brick wall opposite the radiant cooling slab. Both steady state heat flux and a diurnal load simulating the potential gains on the radiant cooling panel were tested.

Steady state performance of the slab indicated the response time of the slab to cooling loads. A continuous load of 354W was generated in the interior of the cube. With water flowing through

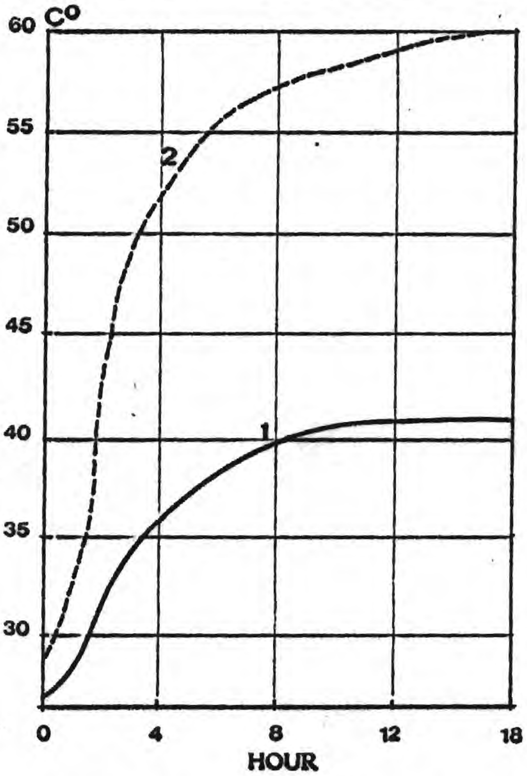

Fig. 3. Steady state profile in calorimetric box with 354W input. Shown are 1: slab surface temperature in front of the pipe, and 2: globe-bulb temperatures in the box.

the slab, 60% of final equilibrium slab surface temperature was reached in 4.5 hours. A total of 13 hours was required to achieve steady state. Temperature profiles of the air and slab surface are illustrated in fig.3. Distribution of temperatures through sections of the 0.05m slab are illustrated in fig.4. A 5.5 C° temperature differential was observed between the slab surface over the pipe, and the slab surface between the pipes. Air temperatures were 19.4°C higher than surface temperatures over the pipe. Temperature gradients within the slab sections are less pronounced at the midpoints between pipes than at pipe sections.

Because applied radiant wall elements will rarely be subjected to steady state loads, intermittent cooling loads are more realistic tests of the performance of radiant elements. The imposed cooling loads illustrate late evening peaks and morning lows and are similiar to those illustrated in fig.1 .

Heat applied to the slab is weighted by a ratio of area of the test slab to the example application This accounted for 1.6% of the buildings total radiant wall areas. This heat flux is doubled to account for spatial concentration of loads within a residence. The panel's surface temperatures follow air temperatures throughout the cooling load cycle as shown in fig.5. Simulations of the radiant panel performance and actual measurements indicate the uniformity of the surface temperatures on the slab. A 1.38°C difference is recorded at times of maximum heat flux. Comparison with steady state performance illustrates the interdependence of the imposed cooling load profile , the radiant wall panel design and the radiant wall surface temperatures. Sections of the slab, shown in fig. 6 illustrate the temperature gradient through the slab with a diurnal cyclical heat input. With this profile heat input to the box, heat flux to the slab varied from 8.52$Wm^{-2}C$ during peak heat input to 7.38$Wm^{-2}C$ at the low point in the morning. These values have been calculated from the temperature measurements and simulations of slab performance on a finite difference model. Surface temperatures of the slab under these conditions were typically 1.4°C above the average water temperature. During peak hours this difference increases to 3.3°C. The surface to ambient temperature difference at these times is 4.4°C. This is well within comfort conditions assuming supply water is 22.8°C rather than 26.6°C as indicated in the tests.

A second series of experiments has been initialized incoporating a solid core brick wall radiantly and convectively coupled to the cooling panel. At present only limited thermal performance data have been recorded for both the steady state and the diurnal performance. These preliminary data indicate the increase in response time of slab surface temperatures and air temperatures to loads. During this test, the water flow rate was increased to $1.06\times10^{-4}m^3s^{-1}$ per square meter of cooling panel area. There was no discernable influence of the incresed flow rate on surface temperatures or response time.

Addition of the brick wall opposite the slab influenced the time period required for the test assembly to reach steady state. Temperatures of both the air and slab at steady state were not significantly different. Temperatures of both the air and slab at steady state were not significantly different from the steady state condition without the brick wall, as shown in fig. 4. Heat flux sensors mounted on the slab for these tests, did not agree with each other. Measurements at surface over the pipe indicated lower heat fluxes (8.52 $Wm^{-2}C$) than at the midpoints between pipes (14.7 $Wm^{-2}C$). A third sensor was used to measure

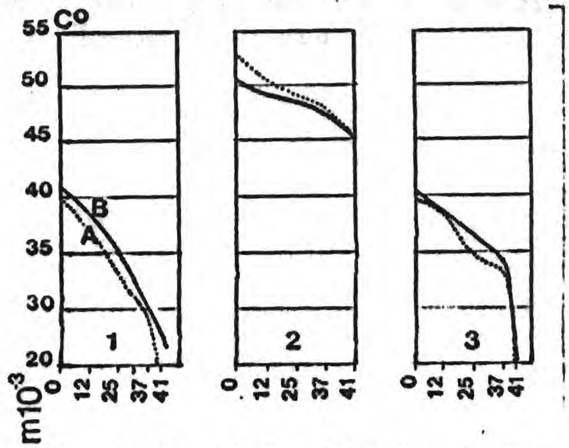

Fig. 4. Sections of the slab and temperature profiles during steady state conditions. 1. over central pipe, 2. over midpoint between pipes and 3. over second pipe. Note the shallow temperature gradient in the midpoint section and the difference between the surface temperatures at various points on the slab. Line A: steady state without secondary mass. Line B: steady state with brick secondary mass.

the convective, radiative split at a point on the surface over a pipe. If compared to meaurements taken at the surface over the central pipe, the convection and radiation coefficients were 4.05 $Wm^{-2}C$ and 4.26 $Wm^{-2}C$ respectively.

Only a qualitative indication of the damping influence of the brick wall has been determined. Brick wall temperatures were recorded 0.6°C below air temperatures indicating depression of the secondary mass' temperature by radiation to the cooling panel.

3. NUTS AND BOLTS

The calorimetric box has proved to be a useful tool for the evaluation of radiant panel thermal performance with steady state and diurnal cooling loads. These preliminary tests have also highlighted numerous pitfalls in the experiments:

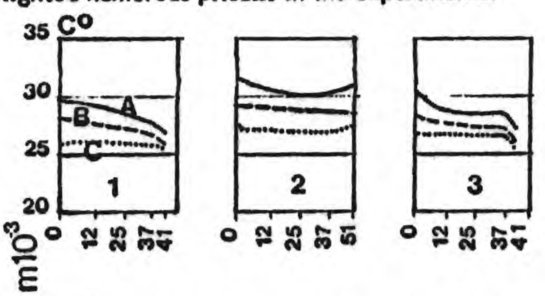

Fig. 6. Sections of the slab and temperature profiles under diurnal cycling. Surface temperature across all three points; 1. central pipe, 2. midpoint between central pipe and a second pipe and 3. second pipe, are close. High tempertures (A) in the slab section occur at 20:00 hrs at time of greatest heat flux. (B) 17:00 hrs and (C) the low occurs at 12:00 hrs.

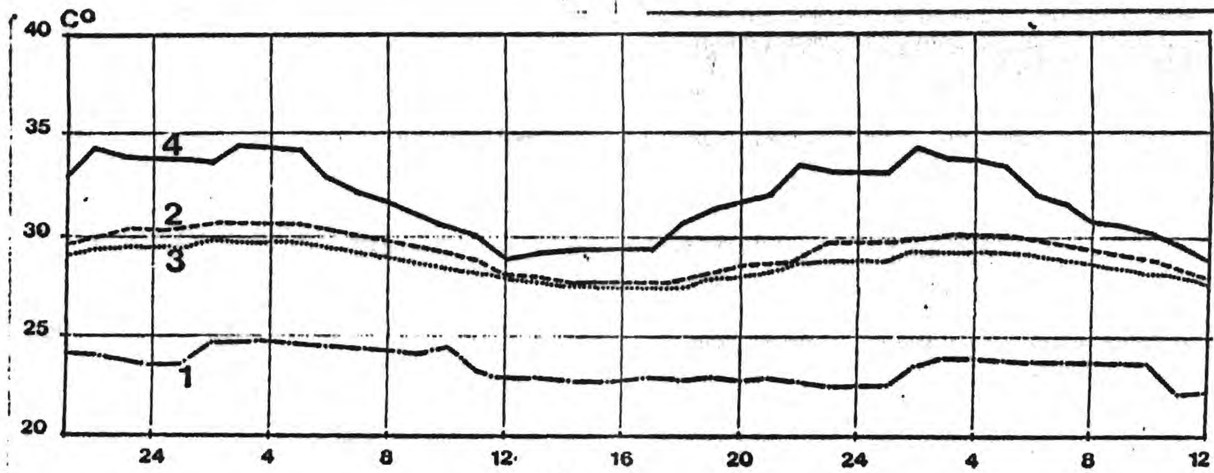

Fig. 5. Calorimetric box performance under diurnal cooling load. The uniform temperature of the slab, is closely coupled with globe-bulb temperature. 1: interior globe bulb temperature, 2:slab surface temperature between pipes, 3: slab surface temperature over pipes, 4:ambient temperature.

- heat flux sensors should be compared under identical conditions before application in differing locations. The author suspects that these sensors are non-linear with increasing temperatures.
- temperature of supply water should be controlled for rigorous replication of applied conditions.
- at the scale of the calorimetric box (a. half scale) convective heat transfer is no longer similiar to full scale conditions. Scaling laws must be developed for rigourous comparison to full scale conditions.

In addition to problems encountered during the experimental phase, the issue of practical applied details must be addressed. Examples of possible applications may be found in ASHRAE's Systems Guide, 1980. A brick cavity wall with copper pipe embedded in the concrete filled cavity is one possible solution. Note that pipe joints within the walls should be avoided by placement of headers in the space above the ceiling joists and below the floor joists.

4. CONCLUSIONS

Few hard conclusions may be drawn from this limited study. Three aspects of radiant cooling panel performance have become clear:

1. the difference in flow rates tested does not influence the panel's response time, nor does it influence the panel's final temperature.
2. radiant coupling to a secondary mass depresses the temperature of the secondary masses and dampens the amplitude of temperature fluctuations in the space.
3. With a fluctuating cooling load profile, a thin slab and wide pipe spacing provides uniform surface temperatures at the slab and maintains comfort conditions in the space.

5. FUTURE WORK

Testing of the high-mass radiant panel's effectiveness as a heat exchanger is continuing with the evaluation of radiant coupling to a secondary mass surface with a diurnal cooling profile. Future tests include the application of drywall to the interior faces of the cube and the evaluation of both steady state and periodic loads on the slab temperatures. As the panel's pipes have valves at both ends of the pipe beyond the slab, evaluations will be performed on the effect of increasing pipe spacing to 0.51m and 0.76m. Performance of a thicker slab (0.102m) will also be examined. Evaluation of cooling panel performance will lead to the incorporation of the most promising designs in a full scale test room.

ACKNOWLEDGEMENT
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INVESTIGATION OF PASSIVE COOLING FOR HOT-HUMID CLIMATES

Progress Report
No. 10

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Heating, and Cooling, Conservation and Solar Applications
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by the

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INVESTIGATION OF PASSIVE COOLING TECHNIQUES

FOR HOT-HUMID CLIMATES

SUMMARY

Work during this report period has been one of frustration. Although much progress has been made, there has been considerable effort directed toward an unsuccessful solution of a field leak problem. The 12'x24' full size radiant cooling panel test room is nearing completion, with test scheduled to begin within two weeks. The heat pump DHW heater has been modified to increase its latent cooling capacity and decrease its sensible cooling capacity through the addition of a run-around coil. Extensive data on the cooling potential of different radiant cooling panels has been developed and is in the process of being put into a form easily used.

As reported in progress report number 9, the cooling field had developed a series of leaks which had been traced to the backfill over the cooling tubes not having been screened. When the first leak was detected, it was decided that the best solution would be to dig up that small part of the field and repair the leak. This was accomplished with some, but not unreasonable, effort. Shortly after the first leak was repaired, a second leak developed. This was repaired and a third leak developed. Repair of leaks continued until eight leaks had been repaired and at least one more remained. At this point it became obvious that the integrity of the field insulation was becoming seriously compromised. Review of the data on hand showed that sufficient data had been taken on the field performance during the previous partial charge and discharge cycle to permit verification of the GROCS and MITAS programs over a complete cycle. Verification of the computer programs' ability to predict accurately the performance of the field over a complete cycle is currently underway.

All leaks have been traced to rocks weighing from 10-500 lbs resting directly on the plastic pipe. The Leaks didn't develop immediately after installation because the failure resulted from plastic flow caused by the rocks resting directly on the pipe over a period of time.

About the time the decision was made to abandon further work on the field because of time and lack of funds to completely replace the field, Georgia Power offered to pay for complete replacement of the field. Georgia Power has been following the progress of the program and is interested in seeing the tests completed.

Delays in getting corporate approval of the field replacement prevented replacing the field in early December as had been planned. Complete approval has now been given and materials obtained. Unfortunately, very cold and wet weather has made it impossible to install the field. The field is now planned for installation a soon as the weather permits the ground to dry sufficiently to get the earth moving equipment into the field. Due to very high interest by people all over the south, the field will be installed and a complete charge-discharge cycle completed next winter, although that is beyond the end of the current DOE contract.

Considerable soul searching and much effort has been directed toward determination of whether others might encounter similar leak problems with the

DET concept. Extensive conversation with Georgia Power personnel regarding installation proceedures for their underground cables has led us to believe that the great quantities of both curbstone and field stone left from an old road which use to pass through the site provided an unusually harsh environment for the field. Georgia Power only cautions their installation personnel to not allow rocks to be put back over the cable for the first foot.

Similar talks with Atlanta Gas Light about their plastic underground gas lines shows slightly greater care. Plastic gas line is made from a special formula medium density polyethylene as opposed to the low density commercial grade polyethylene used in our original field. the pipe has a .150" wall thickness rather than the .108" wall thickness of the pipe used in the original field. Fortunately, the 2306 polyethylene gas pipe is readily available and is only slightly more expensive than the general purpose polyethylene. The plastic gas lines are backfilled with a screened soil for one foot before soil removed from the hole in which the pipe is installed is returned to the hole.

Polyethylene 2306 gas pipe will be used for the new field. Figures 1 and 2 are photographs of the pipe to be used, showing the greater wall thickness and the standard to which the pipe is manufactured. A pipe of 1.25" nominal inside diameter will be use rather than the 1.5" nominal inside diameter used in the original field because it is more readily available in the Atlanta area.

Although the field leak problem has caused many wasted hours and the loss of much data regarding the performance of the cooling field through a complete charge-discharge cycle, it may be a blessing in disguise. Our initial field is an almost exact copy of the fields used by Brookhaven National Laboratories in their ground coupled water source heat pump studies including the use of low density polyethylene tubing and the method of installation. There have been no leaks reported in their fields. If we had installed our field in a site with no rocks, we probably would not have developed leaks and would have come to the conclusion that leaks were not a significant problem. Since we did develop leaks, despite what could be considered a reasonable effort to minimize problems with leaks, we have been forced to look at the problem in some detail and to develop installation methods which should work in all soils.

Replacement of the field permits us to make several significant improvements in the field which will increase performance and reduce cost. Our computer simulations have shown that even with the 2" of insulation over the field, significant energy is lost through the insulation to the ground surface. It would be desirable to minimize field plan area, and thus surface area through which energy can escape. The companies who have estimated the installation cost of the field all agree that a 30' X 40' hole 8' deep is no greater in cost than one 60' x 40' that is 4' deep. The new field will have the smaller plan area and will be 8' deep with a two level coil plane. Figure 3 shows an exploded view of the new field, while Figure 4 shows a cross section of the field as it will be installed.

Use of the 1.25" pipe permits us to go to 3' spacing between the coils and to get 1000' of pipe in an area occupied by 700' in the original field. The pipe will come from the building, go to the air-water heat exchanger and then drop to the 8' depth. It will go to the lower edge of the field and begin to serpentine back toward the exchanger with each tube spaced 3' from the adjacent tubes. Once an area 30' X 40' has been covered, the tube will turn upward to the 4' level where a second plane will duplicate the 8' plane. Once the plane at the 4' level

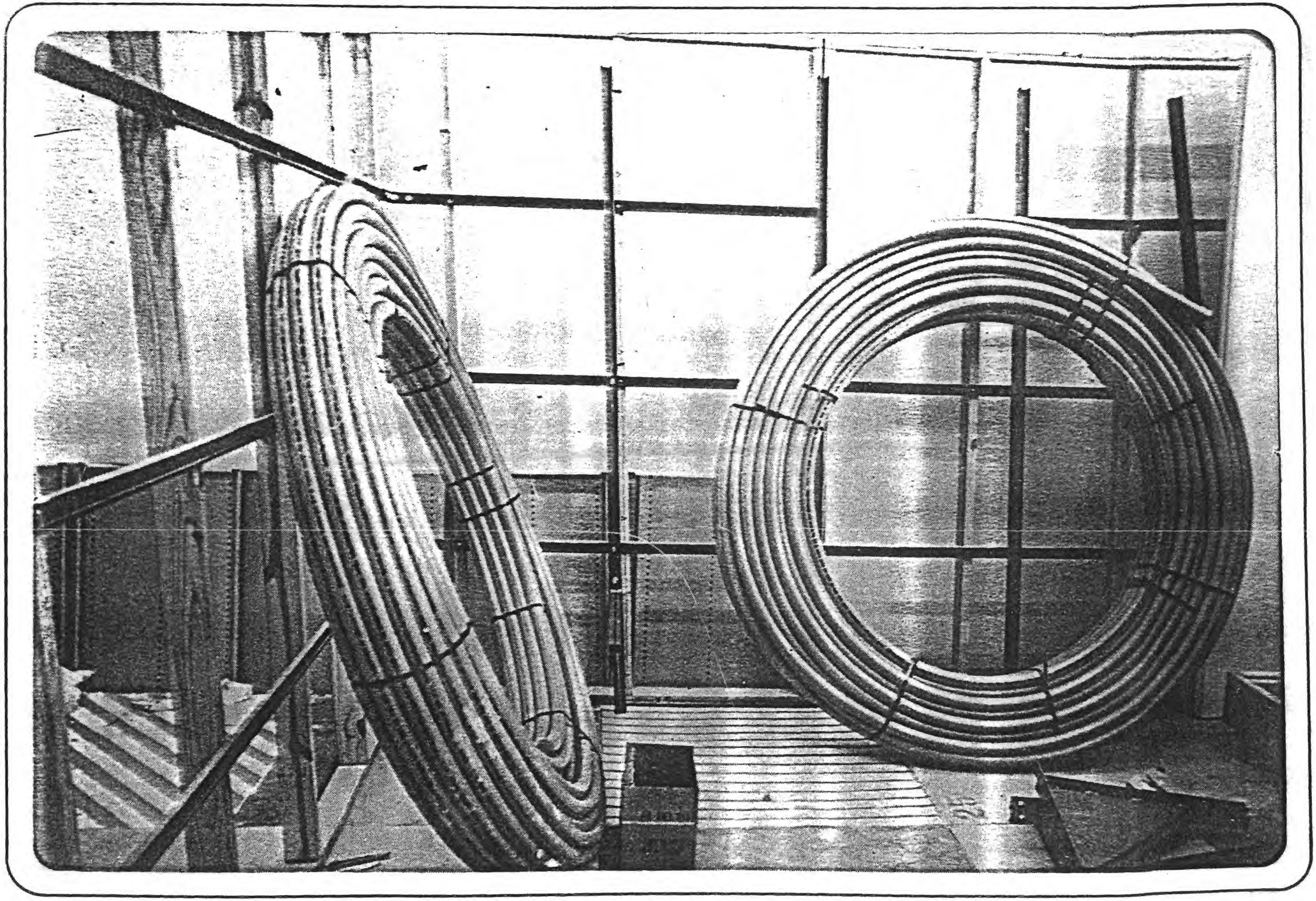


FIGURE 1   PHOTOGRAPH OF POLYETHYLENE 2306 TO BE USED IN NEW FIELD

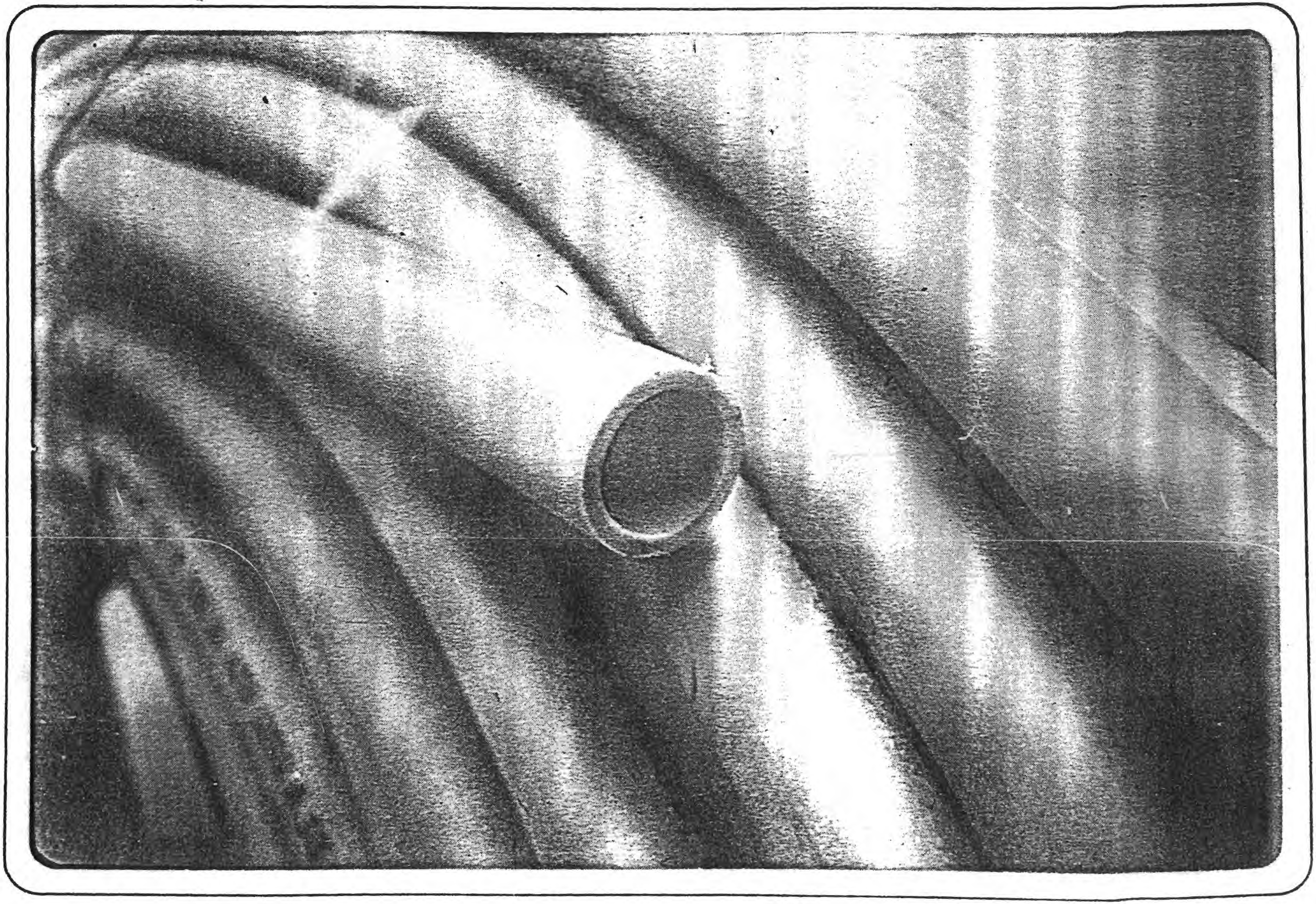
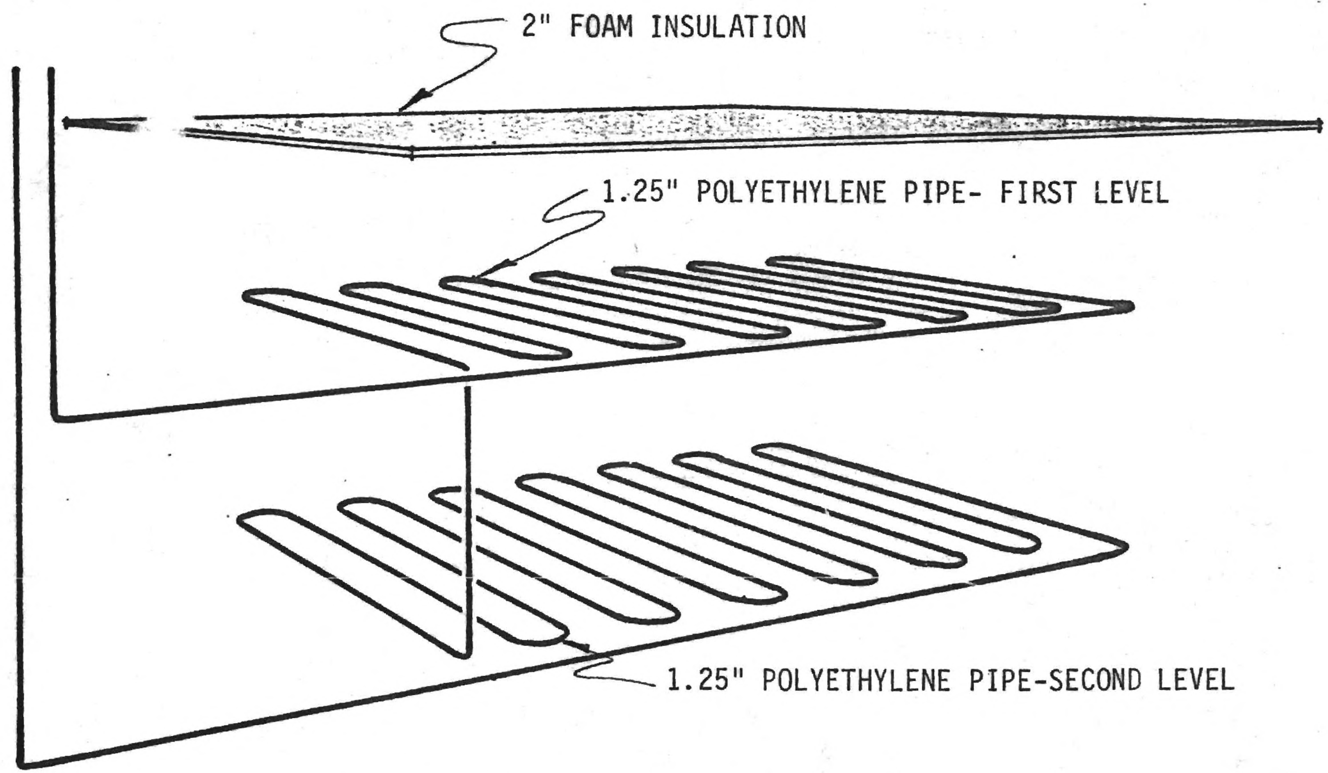


FIGURE 2 PHOTOGRAPH SHOWING INCREASE PIPE WALL THICKNESS



EXPLODED VIEW OF TWO LEVEL COOLING FIELD

FIGURE 3   EXPLODED VIEW OF TWO LEVEL COOLING FIELD

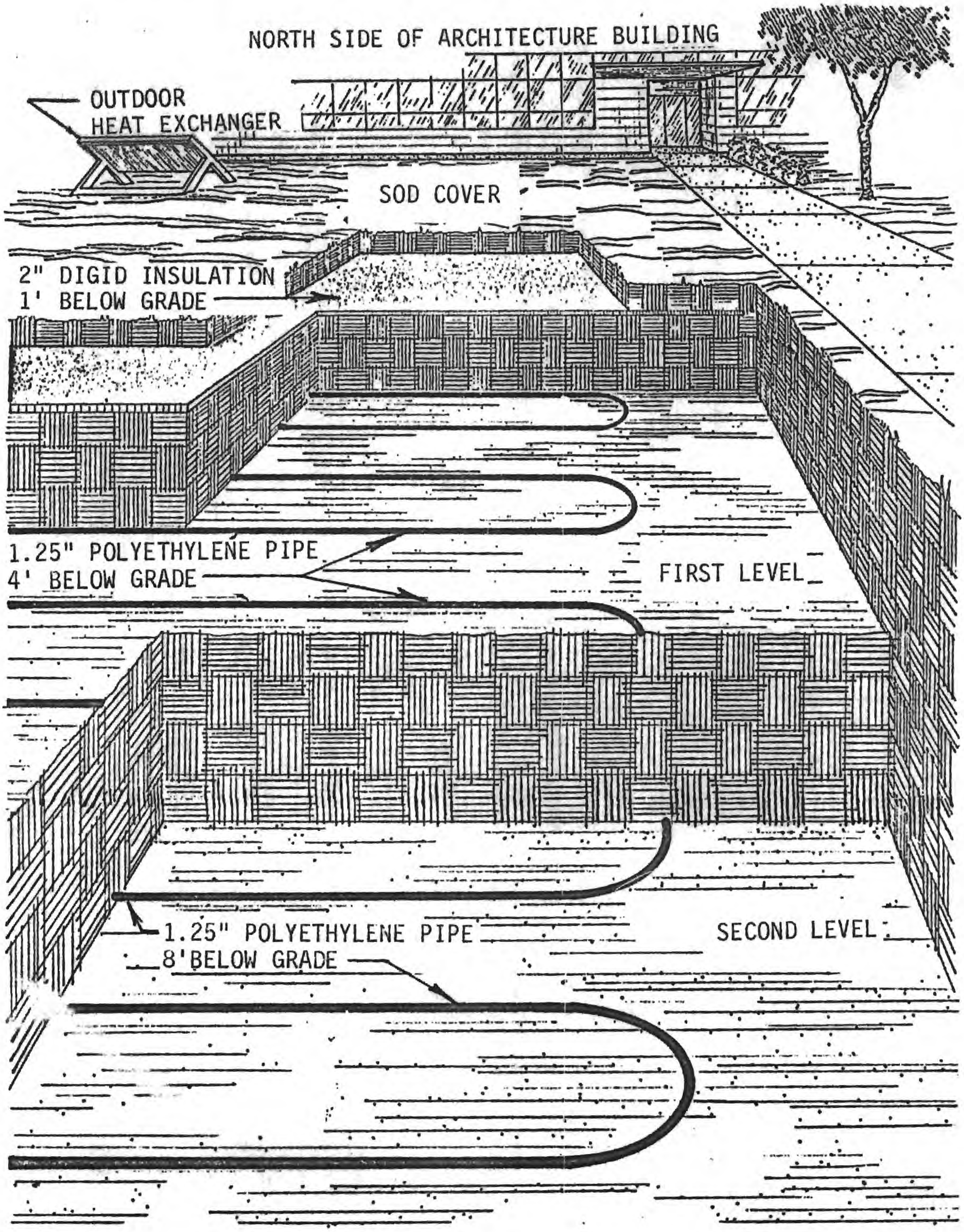


FIGURE 4 ISOMETRIC OF NEW FIELD

has been completed, the pipe will return to the building. With this arrangement, the surface area that must be insulated will be reduced about 40%, thus reducing one of the more costly items in the field. Insulation will extend 8' beyond the edge of the deeper field as opposed to the 4' extension used on the 4' deep field. With less plan area, energy loss to the surface will be significantly less. The 30' X 40' area is now small enough to fit under most reasonably sized houses, where its performance will increase further and cost reduced further because losses through the surface will be to the house. Since most of the losses occur in early summer, they will not be losses when under a residence because they will assist in cooling the residence directly.

Test Room

Work on the 12' x 24' radiant panel test room is complete. Figures 5 and 6 show the test room. Performance test using this test room should begin within two weeks. Figure 7 show the heat pump and water storage tank that will be used to supply chilled water to the test room.

The cooling pipes running through the wall in the test room are biased toward one side so that the wall can be used to evaluate two different pipe depths, either individually or simultaneously. Individual evaluation only requires insulation of the wall side not being evaluated. One of the rooms is presently being planned for use as an instrumentation room for the test wall.

Humidity Control

Progress report number 9 pointed out the necessity of carrying the latent load with a mechanical system and the desire to minimized the sensible load carried by the mechanical system. A heat pump DHW heater was suggested as an ideal device for accomplishing dehumidification because of the necessity to heat the water anyway. ...e 8 and 9 shows a commercially avaliable heat pump DHW heater. Figu... shows those components necessary to increase the latent capacity of th... dev...e and decrease the sensible capacity. Notice that only two readily availa... ai... to water coils and one small pump with two short hose connections are ...c...sary to add the run-around coil to the unit. Figure 10 shows those components ...ey are installed on the unit.

Modification o... he heat pump DHW heater is nearing completion. Initial tests should be conducte... within a couple of weeks. Critical tests cannot be easily conducted until the summe... months when the humidity is high.

Radiant Panel Test Box

Considerable test...ng has ...directed toward determining the radiative cooling potential of n...merc... ... radiant cooling test panels using the radiant panel test box. Figures 11-15 summarize the performance of five different test panel configurations. One can see the improvement in performance under a transient load en...vi...nent when mass is added to the radiant panel. All of the panels have b...n t...d under both steady state and transient load conditions. Since stea...y stat... temperatures are only dependent upon the inside-to-outside temperatur... and the thermal resistance of the walls, we will concentrate here on th... transient load analysis.

Figure 11 show... the load profile used in all of the tests. Figure 11 also shows the variation ... temperature of four different locations for a 2" half

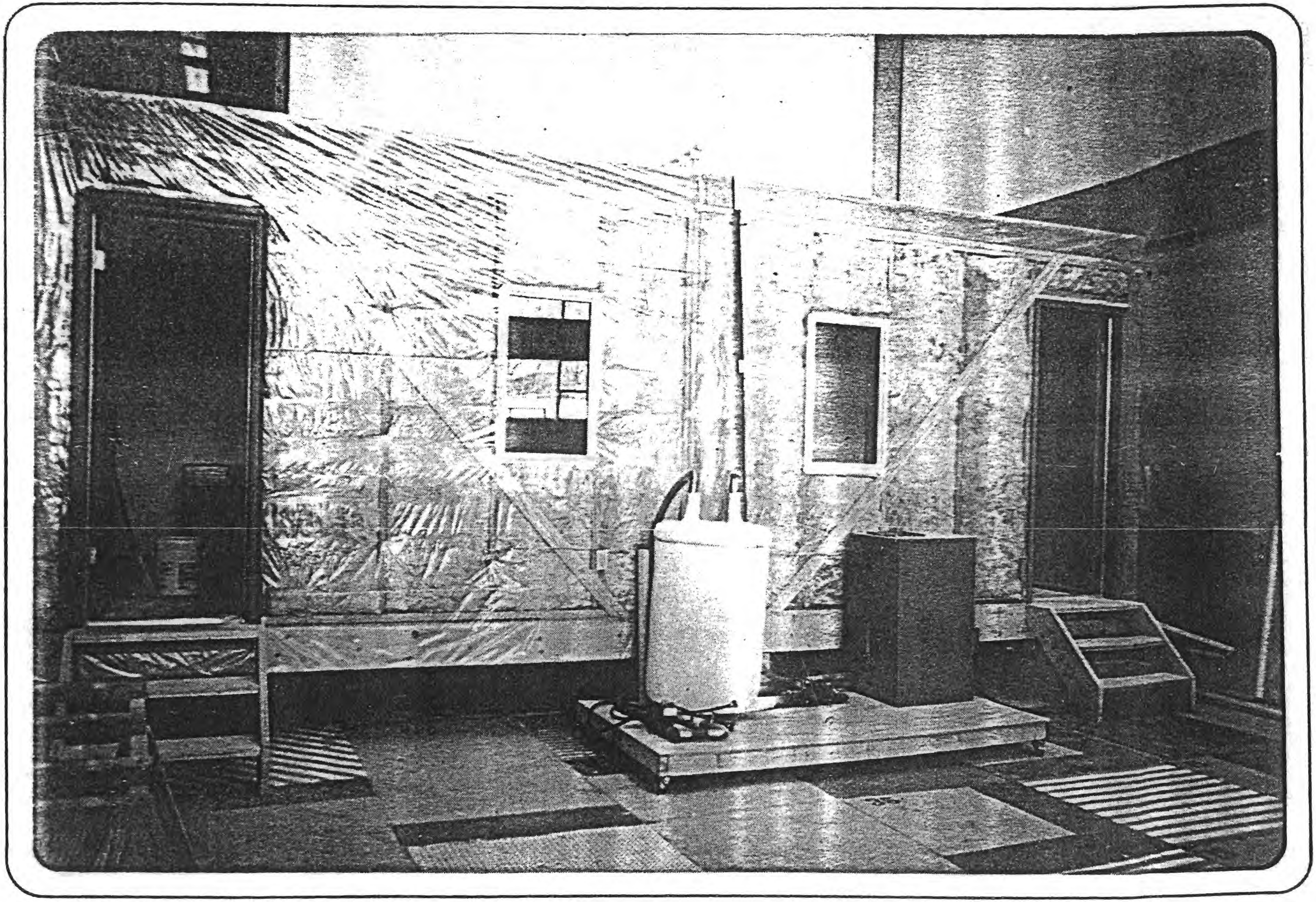


FIGURE 5 PHOTOGRAPH OF FULL SIZE TEST ROOM

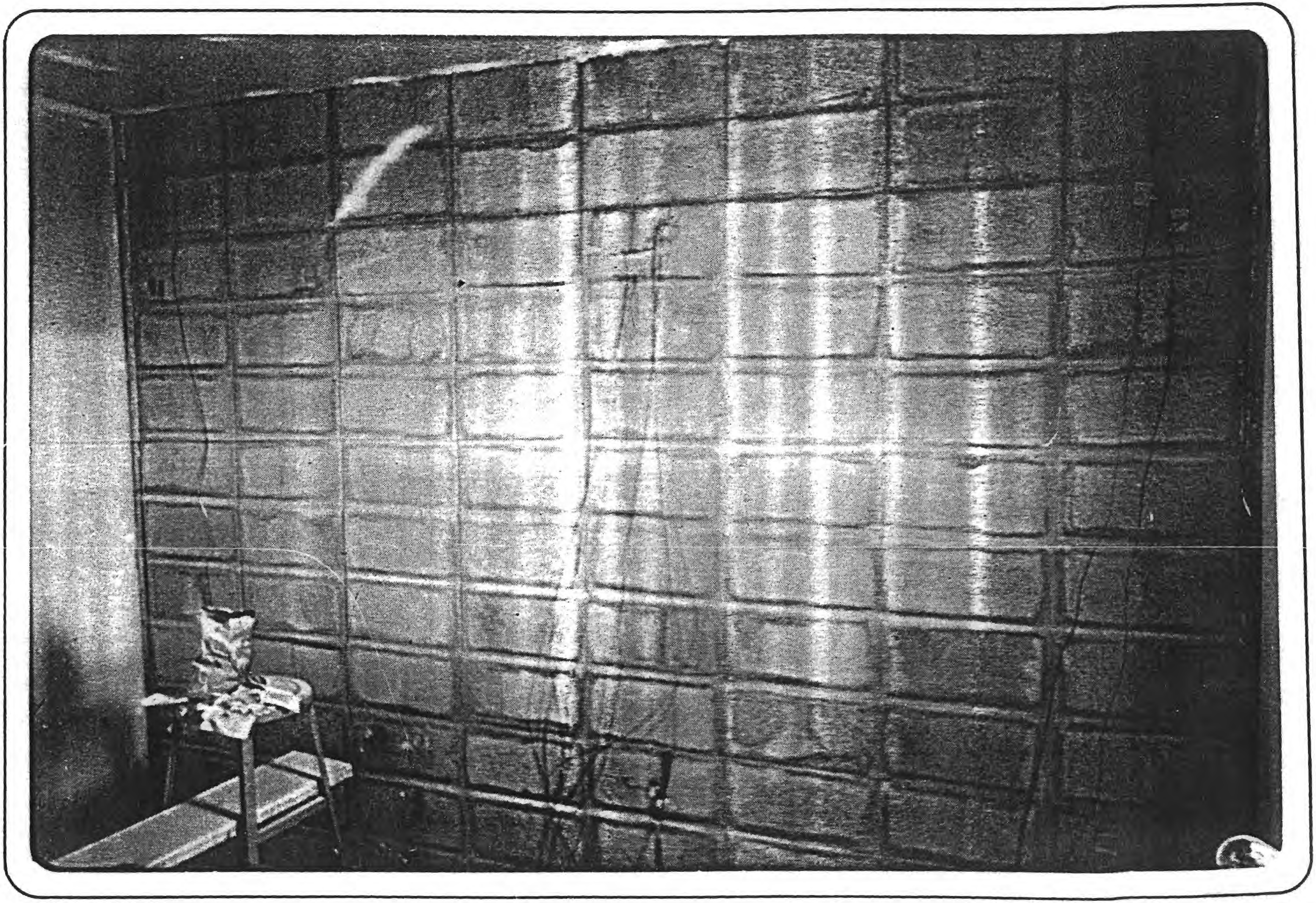


FIGURE 6   PHOTOGRAPH OF FULL SIZE TEST WALL

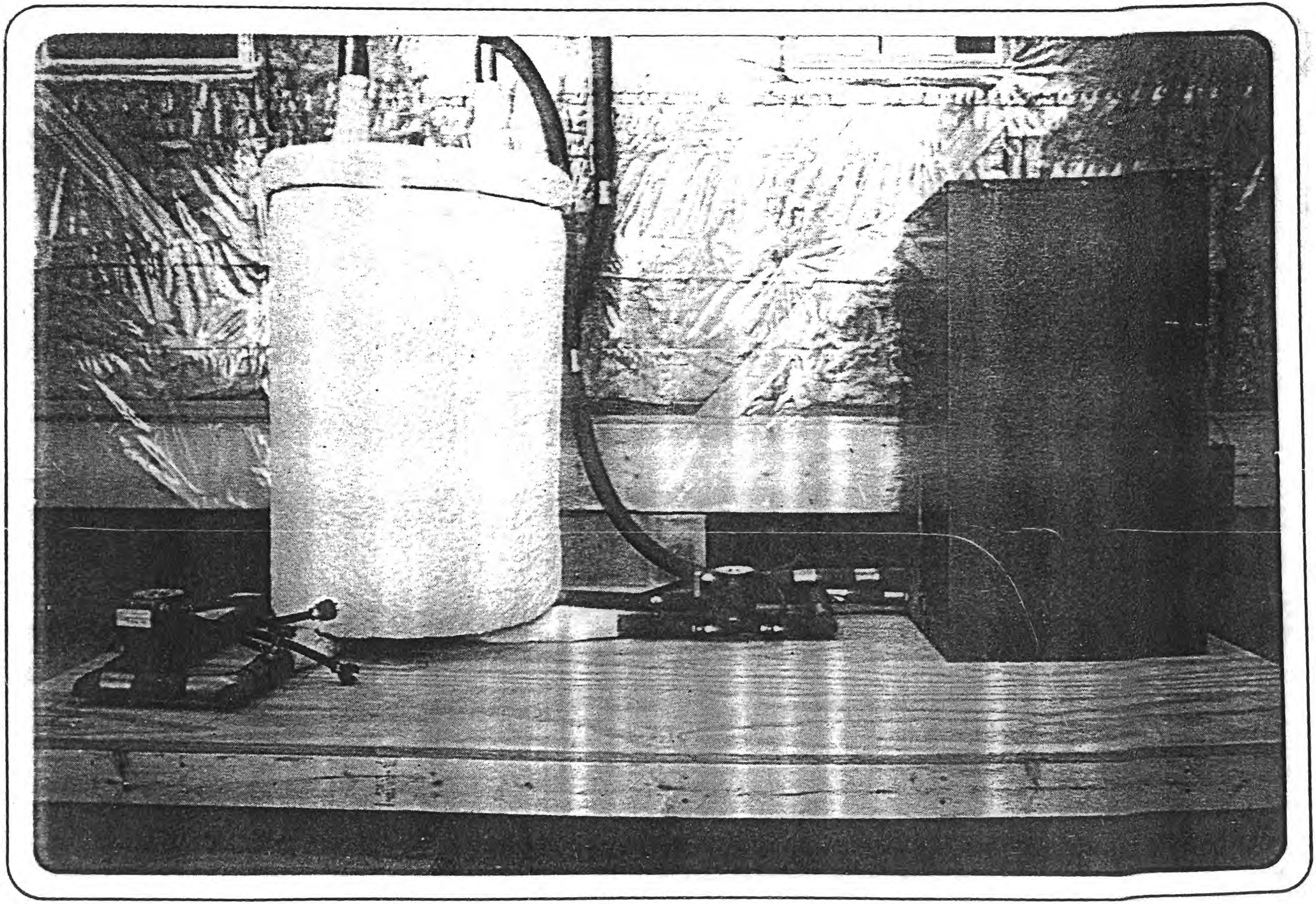


FIGURE 7 HEAT PUMP AND WATER STORAGE TANK

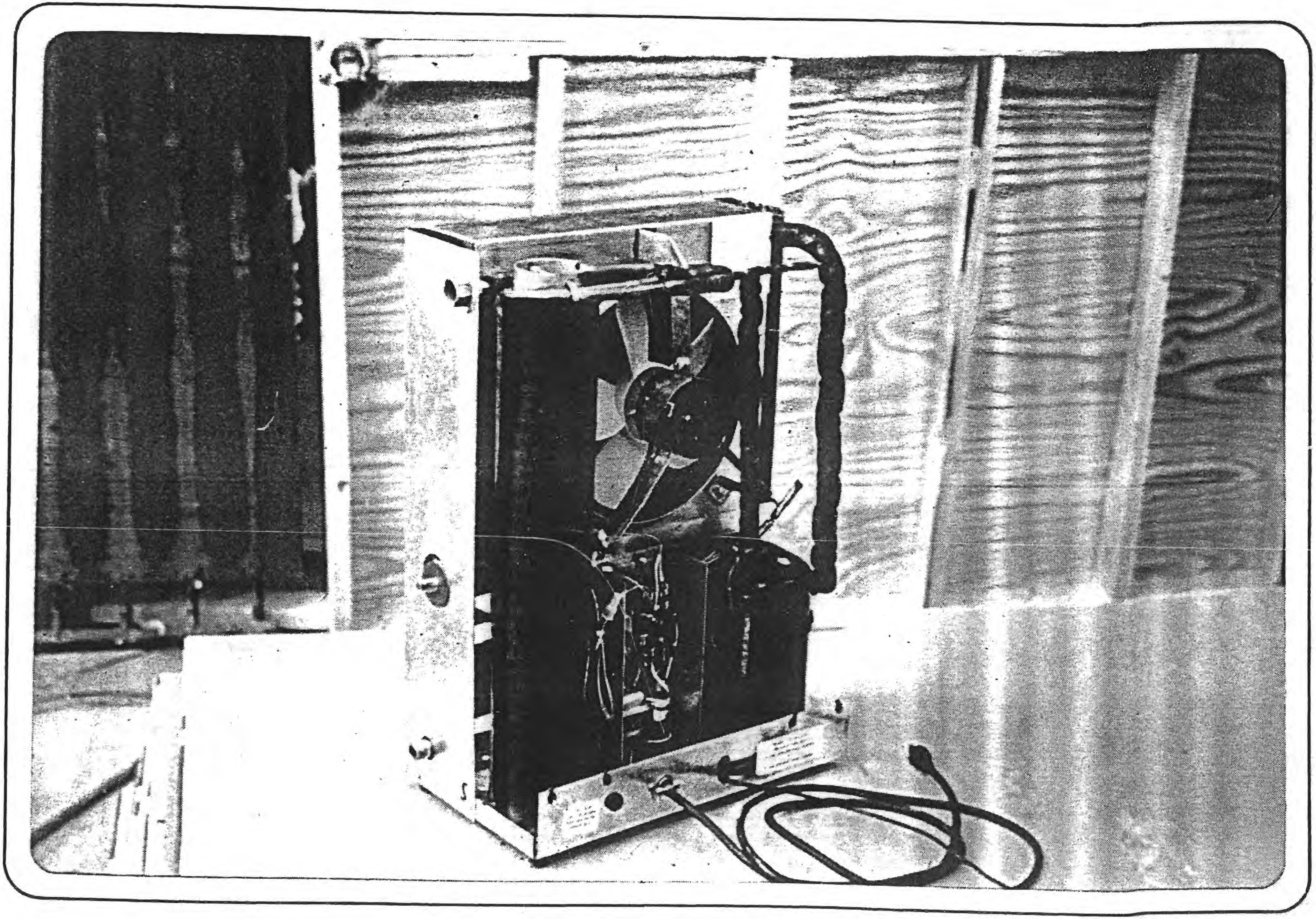


FIGURE 8   PHOTOGRAPH OF ETECH DHW HEATER WITH COVER REMOVED

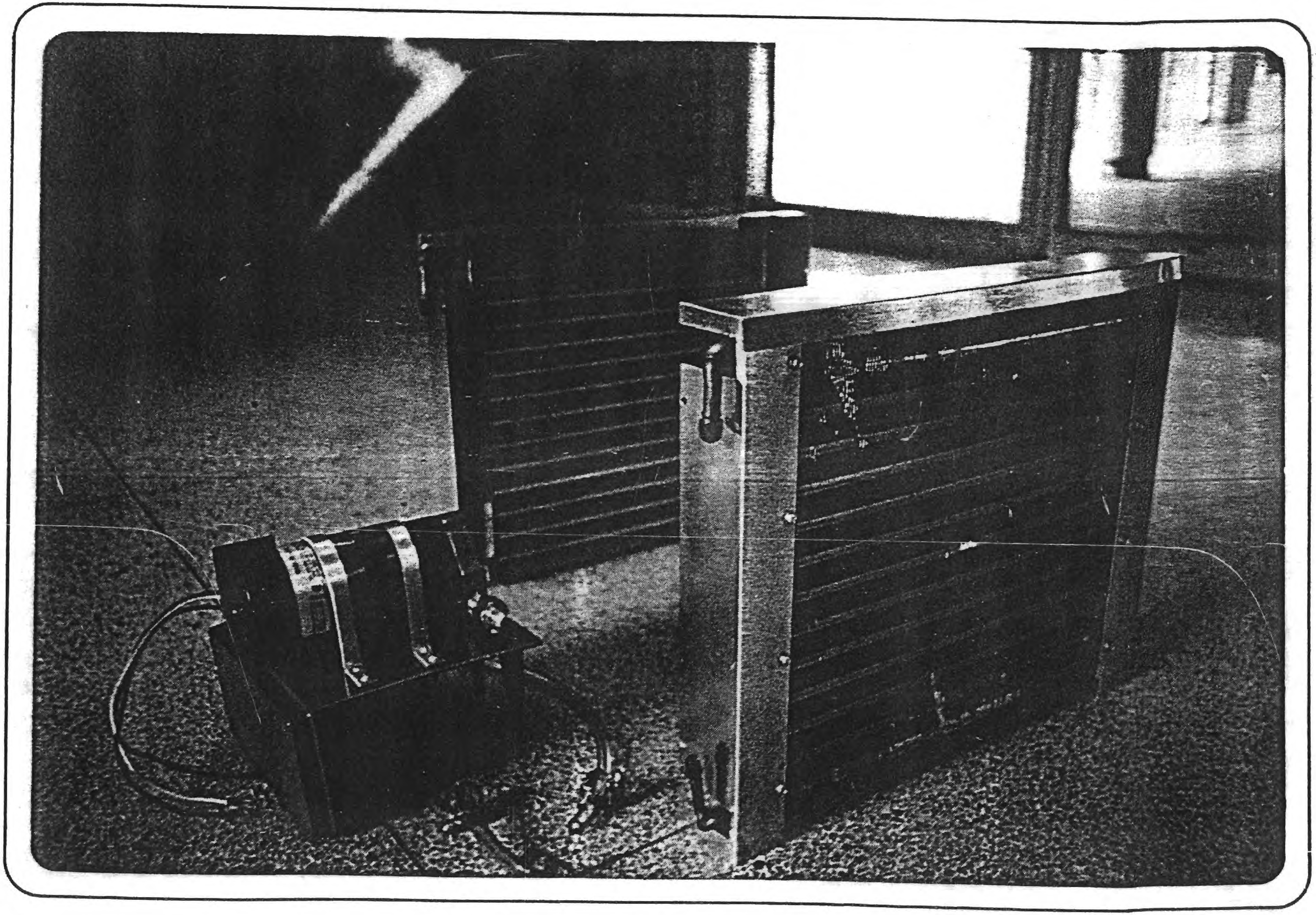


FIGURE 9 PHOTOGRAPH OF RUN-AROUND CYCLE COMPONENTS

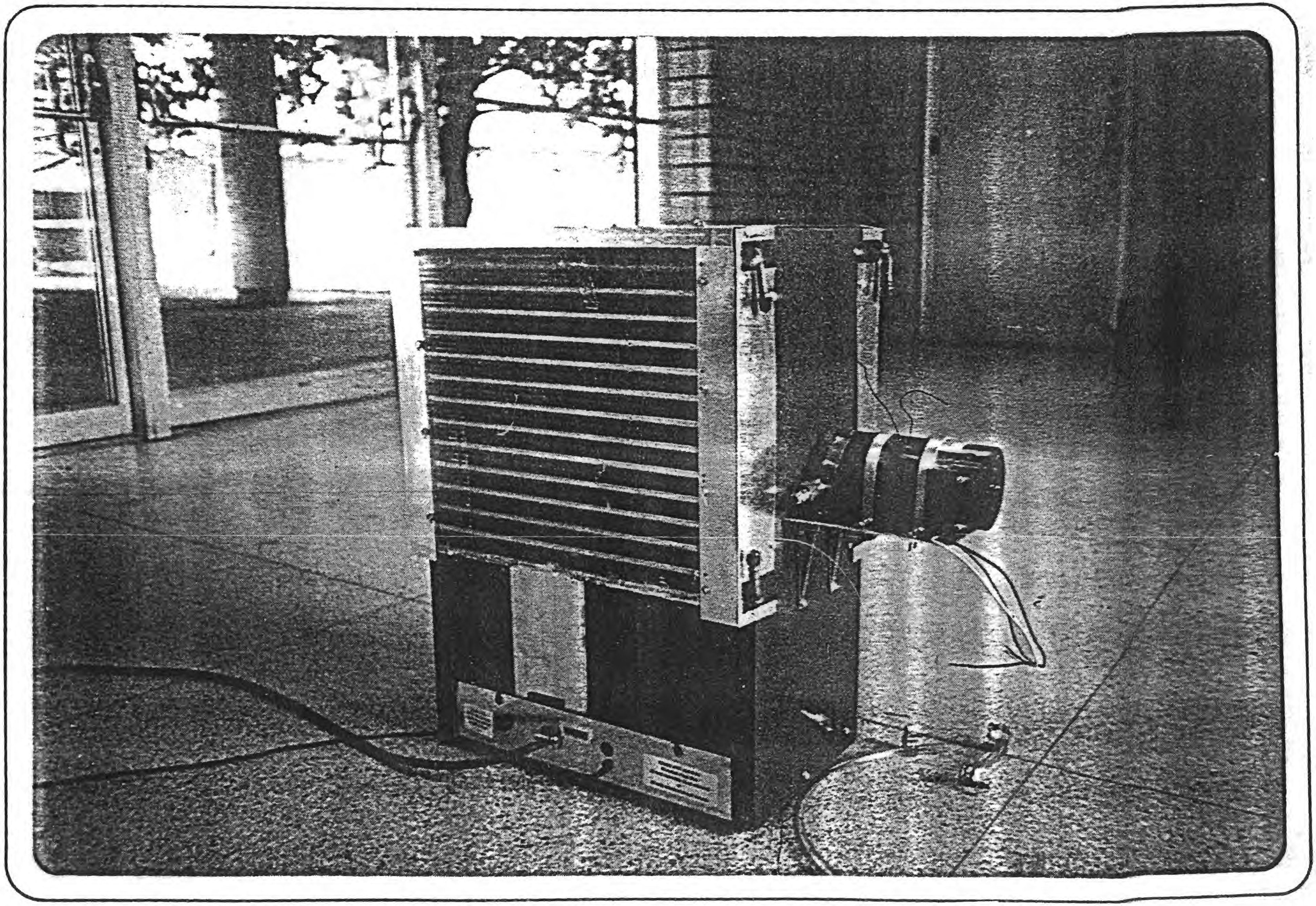


FIGURE 10 RUN-AROUND COIL ASSEMBLED ON DHW HEAT PUMP

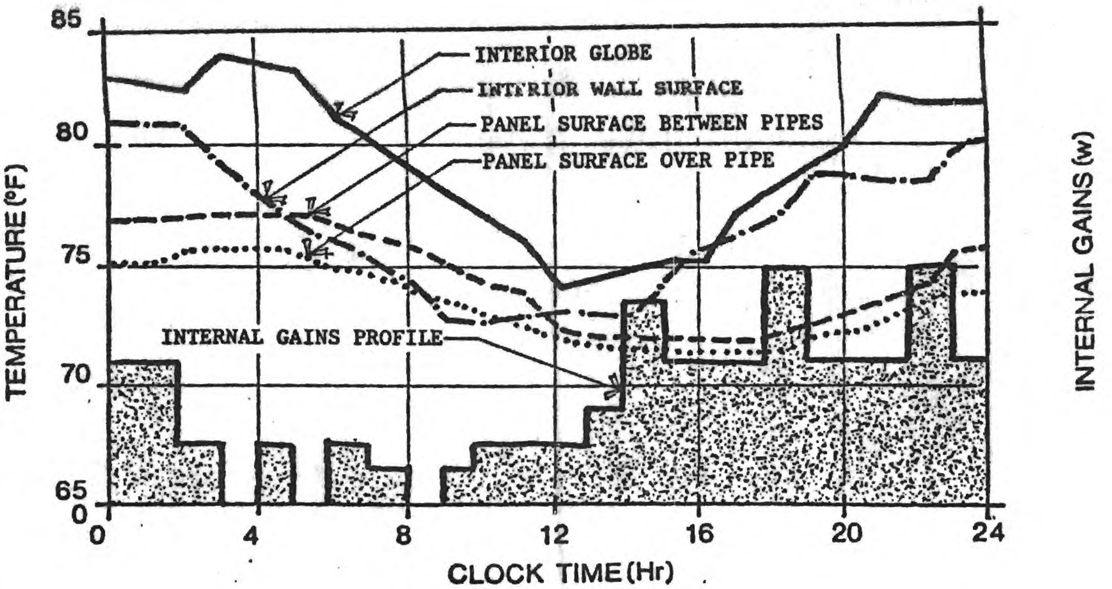


Figure 11. Thermal Performance of 2" (Half Thickness) Radiant Cooling Panel with 10" Tube Spacing

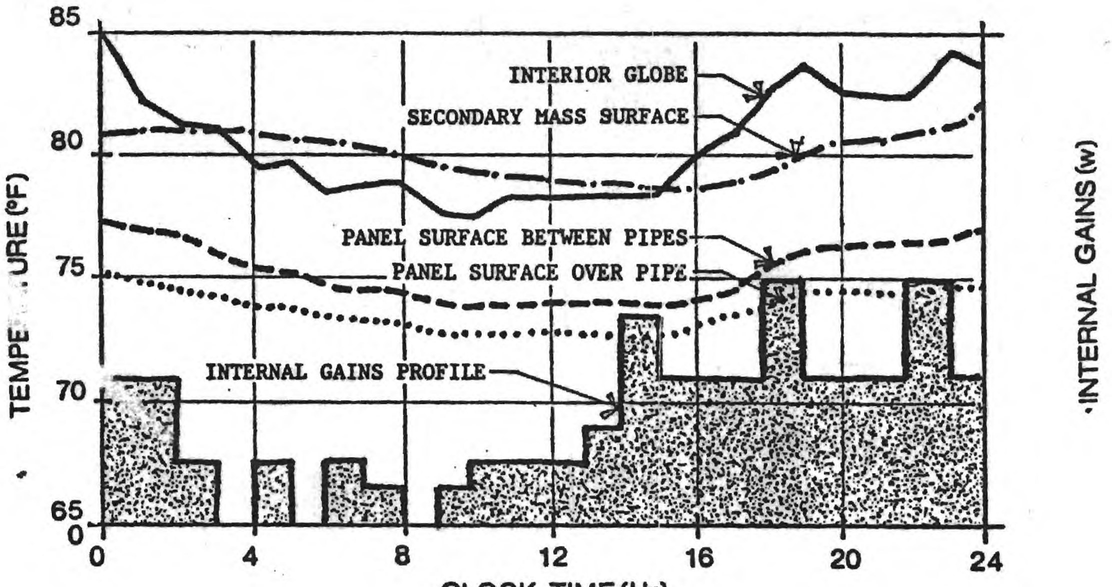


Figure 12. Thermal Performance of 2" Radiant Cooling Panel with Secondary Mass Wall, 10" Tube Spacing

thickness wall with water tubes spaced 10" apart. Figure 12 shows the same test except with a brick wall added to the test box directly opposite the radiant cooling wall. Notice that all of the temperatures fluctuate much less when the additional mass is added. Figures 13 and 14 further illustrate the importance of secondary mass on the performance of the radiative cooling panel. Figure 13 shows how the additional mass decreases the temperature fluctuations of the panel directly over the tubes, while Figure 14 shows a similar effect midway between the tubes. To facilitate the test schedule, thicker radiant panel testing has been scheduled later. Test of these thicker panels are just not beginning.

It had been suggested by several people interested in the program that performance of the system could be greatly improved by using a radiant panel with a high conductivity, similar to the metal panels used in radiant heated ceilings. Figure 15 shows the assumption of improved performance with increased conductivity is not valid if one must sacrifice mass as conduction area for conductivity. The steel panel used in the tests whose results are shown in figure 15 was .062" thick with copper tubes bonded with high conductivity epoxy every 10" (the same tubes and spacing used with the concrete panels whose performance is shown in figures 11-14). With water at a fixed temperature and flow rate, the steel panel is now able to conduct energy to the area directly over the tubes fast enough to meet the instantaneous loads. With very little mass the steel panel could not store cooling capacity as does the concrete panels. One can't disagree with the premise that high conductivity is desirable. One can disagree if one must sacrifice mass to obtain high conductivity. Since the field studies have already shown that the field could probably meet the average daily load, but might be incapable of meeting the instantaneous peak loads these tests are reassuring.

Future Work

Work during the next report period will concentrate on completion of the full size test room and initiation of testing full size panels using the new radiant test room. Test will also continue on different radiant test panels using the radiant test box. The modified heat pump DHW will be completed and initial test conducted. The field will be replaced using the new configuration as soon as the weather will permit the earth moving equipment to work efficiently. The computer programs will be used to predict the performance of both a single and double plane cooling field based on its agreement with the performance of the single plane field last season.

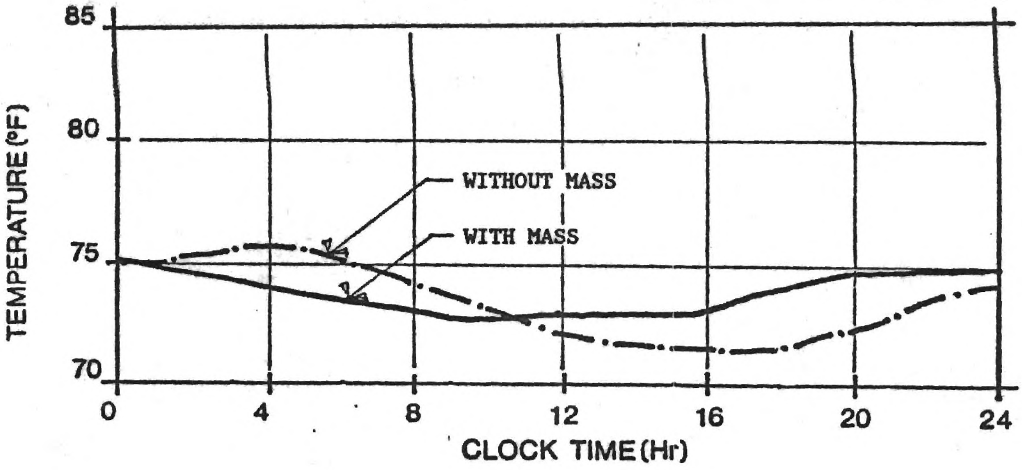


Figure 13. Panel Surface Temperature Directly Over Tubes when Subjected to a Dynamic Load, 10" Tube Spacing

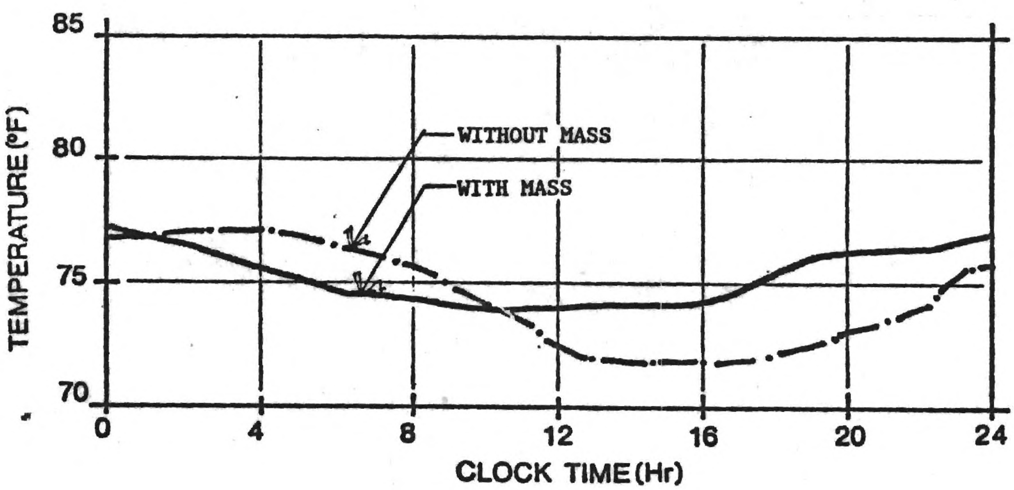


Figure 14 Panel Surface Temperature Midway Between Tubes when Subjected to a Dynamic Load, 10" Tube Spacing

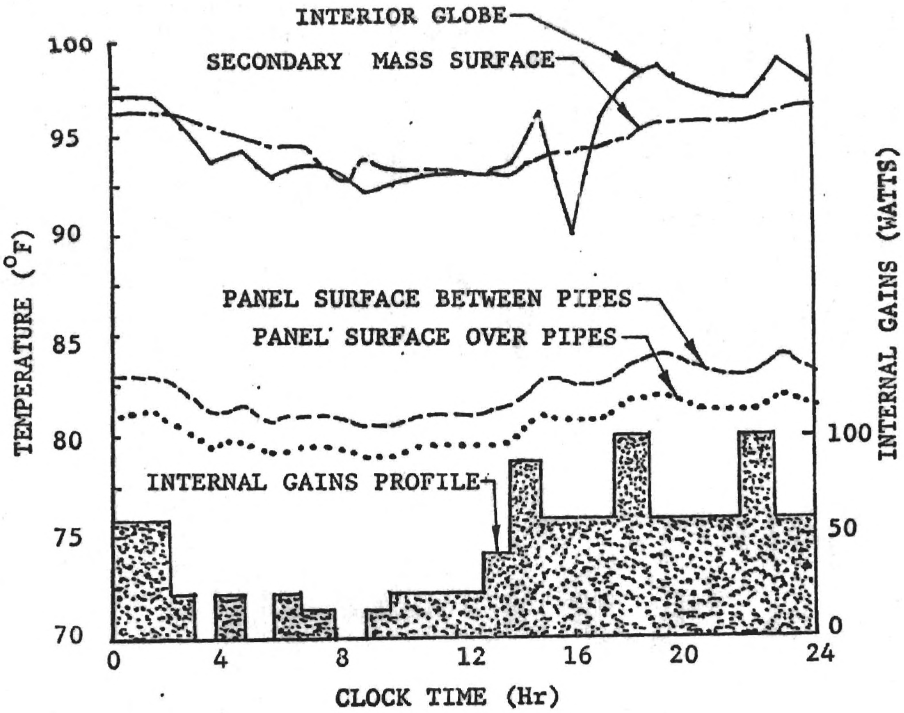


Figure 15 Thermal Performance of a .062" Thick Steel Radiant Cooling Panel with 10" Tube Spacing

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FINAL REPORT

INVESTIGATION OF PASSIVE COOLING TECHNIQUES FOR HOT-HUMID CLIMATES

By
James M. Akridge

Prepared for

The U.S. Department of Energy

Under

Contract No. DE-AC02-79CS30238

October, 1982

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ATLANTA, GEORGIA 30332

1982

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The author wishes to express his appreciation to the Department of Energy for funding this program and providing technical guidance throughout the program. The author would also like to thank the Georgia Power Company for their financial support in replacing the cooling field.

CHAPTER I

BACKGROUND

As energy has become more and more scarce and costs have continued to rise, it has become imperative that all segments of the American society undertake significant measures to reduce energy consumption. Approximately thirty-five percent of the energy consumed in the United States goes to heat and cool buildings.[1] Significant reductions in energy consumption are possible through changes in the way buildings are operated (energy management). Further reductions are possible through the use of better and more efficient mechanical systems, and through the use of active solar heating and cooling systems, although the latter is questionable from an economic sense at this time.

It has become increasingly obvious that the greatest potential for energy reduction lies in the proper design of the buildings. Passive architecture, as it is popularly called, is not new. Before the advent of efficient mechanical systems and readily available fuels, passive architecture was the primary means one had for maintaining comfort, other than through the use of clothing and localized fires.

With cheap energy and mechanical systems which are capable of completely controlling an environment, designers began to rely less and less on the building itself to control its environment. This has led to the design of buildings over the last forty years which are less efficient each year, i.e., use more energy per $ft.^2$, than the buildings designed the previous year.

As fuel costs began to rise in recent years several designers began to design buildings to be passively heated. This concept of using the building itself as a solar collector has received considerable attention and can be accomplished through proper design in most sections of the United States. More and more buildings are being designed to be passively heated. Many architects and designers have learned how to incorporate passive heating features into both residential, commercial and industrial building. Researchers like Balcolm[2], Mazria[3], TEA[4], and many others have developed design guideline for the design of passively heated buildings.

Although many of the same principles useful in passively heated buildings are also applicable to passively cooled buildings, passive cooling has proved to be much more difficult to accomplish effectively. This is particularly true in hot-humid climates. Where nighttime temperatures are relatively low and/or atmospheric moisture content is low; diurnal, evaporative cooling and radiation to the sky have all proven to be effective methods for cooling buildings.

Unfortunately, much of the United States which has high cooling loads lies in what is popularly called a hot-humid region. Direct evaporative cooling will not provide an increase in comfort during much of the cooling season due to relative humidities which are already higher than desirable. Radiation to clear night skies is also much less effective due to moisture in the air acting as an infrared trap.

LITERATURE SEARCH

One of the initial tasks of this program was to thoroughly search the literature to determine whether there are cooling techniques which have been used and which are effective in reducing cooling loads in hot-humid regions. The literature search did not find any passive cooling techniques that these researchers were not aware of before the search was started. It also did not find any existing passive cooling techniques which show promise for use by the majority of the building industry. Much of the more promising techniques are the obvious ones, such as the use of shading, light colored roofs, and walls, insulation, water sprayed roofs, etc. Most of these were investigated thirty years ago in Israel, South Africa and in the United States. Thirty years ago fuel was cheap and mechanical systems were much more convenient and effective so most of these techniques were not pursued or developed further until recently.

The literature search also revealed that many poorly qualified "authors" are extolling the merits of various passive cooling techniques without either theoretical or experimental verification of their claims. Other authors have presented papers with titles which suggest that methods for passive cooling in hot-humid climates were being presented. Unfortunately, a thorough review of these papers reveals that the papers' authors defined, frequently very eloquently and thoroughly, the problems with passive cooling in hot-humid climates without offering any solutions. These authors have at least recognized the problem and are not offering hope where hope may not lie.

Another group of papers fall into what possibly could be called promising, but needing additional work. Many of the earth cooling tube concepts fall into this category. The idea may have merit but it also has problems which have either not been recognized or have been ignored. Most proponents of this system have ignored the problem of air being discharged into a living space at 70-75^{o} F in a saturated state. Even when mixed with dry indoor air, comfort is not likely to result. Several papers suggest computer simulations show that ground conductivity is not really important and that thermal saturation is not likely to occur. This area needs considerable additional work. Experience with ground loop heat pumps 20-30 years ago indicates that ground saturation may be a problem.

Earth tubes would appear to have some promise in certain soils (soils with high moisture content, for example) especially when using a misting chamber on the inlet to the earth tube in dry climates. One problem which must still be addressed is that also found with the use of diurnal rock beds in hot-humid climates. Conditions in the earth tube or rock bed are ideal for growth of bacteria, fungus, mold and mildew. Conditions close to those have been found to be growth centers for the Legionaries Disease.

In summary, one can say that no passive cooling techniques exist which show promise of being effective in hot-humid climates. There have been many papers and reports written claiming to show effective techniques. These claims were either not presented or were not based on scientific grounds and did not deliver what they promised.

The following chapter briefly describes each feasible passive cooling technique and describes its application and limitation when applied to hot-humid climates.

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CHAPTER II

POPULAR PASSIVE COOLING TECHNIQUES

Numerous systems are being presented in the literature as being viable passive cooling techniques, frequently with little or no qualification regarding the type of climate in which they are applicable. Since most of these have significant limitations and are not applicable to hot-humid climates, it was felt appropriate to briefly discuss a number of these and indicate their limitations and potentials.

1.0 LOAD CONTROL

All approaches to passive design whether heating or cooling, must start with the premise that the structure is designed to minimize loads due to external conditions. This means that insulation in walls and roofs have been optimized, that the building has been properly oriented to take advantage of prevailing winds and the sun, that windows have been properly located, sized and shaded and that optimum use has been made of local terrain and trees. Passive cooling design also requires the use of light colored roofs and walls where possible. While the above might be considered passive cooling concepts, they are more appropriately called load minimization techniques. They are useful and desirable whether the building is passive or not.

2.0 VENTILATION

2.1. Natural Ventilation

This is a time-proven concept and was the primary passive cooling technique used in the southeast before the advent of electric fans. It is a very effective passive cooling technique for climates where the maximum daily temperature remains below

32.2°C (90°F) and where the relative humidity remains low. Within limitations, increased air velocity will result in an increased level of comfort. Designing for natural ventilation simply means designing a structure so that air movement through the structure is high. This means orienting the structure so that windows face into the prevailing wind during those periods when cooling is desired. It also means proper location of windows so that prevailing winds will flow through the structure rather than around the structure. Flow through the structure must be through occupied zones. When temperature and relative humidity is moderate, as in the spring and fall in the southeast, ventilation is an effective cooling method.

When temperature and humidity rise to levels typical of the southeast during the summer months, ventilation is seriously limited in its ability to provide an increased level of comfort, especially during the day-time hours when both temperature and humidity is high. Figure 2.1 taken from Givoni[1] shows that increased air velocity becomes ineffective as the air temperature approaches 32.2°C (90°F). It also shows that as relative humidity rises, the air loses its ability to evaporate moisture from the human body, thus becoming quite ineffective as a cooling media. While Givoni showed increased comfort with moving air at relative humidity approaching 100%, experience has shown humidity above 70% to be prone to mold and mildew problems.

2.2 Thermally Induced Ventilation

The "Solar Chimney" has received much attention and is being included in a number of houses presently being built in the southeast. Basically, a solar chimney depends upon solar energy to heat the air in a vertical column. As the air becomes heated, it rises, pulling ambient air through open windows and doors. Thermally induced ventilation has several serious limitations. Abrams[2] and Brock[3] independently showed that wind velocities of 3-5 mph are sufficient to totally overpower the thermal effects

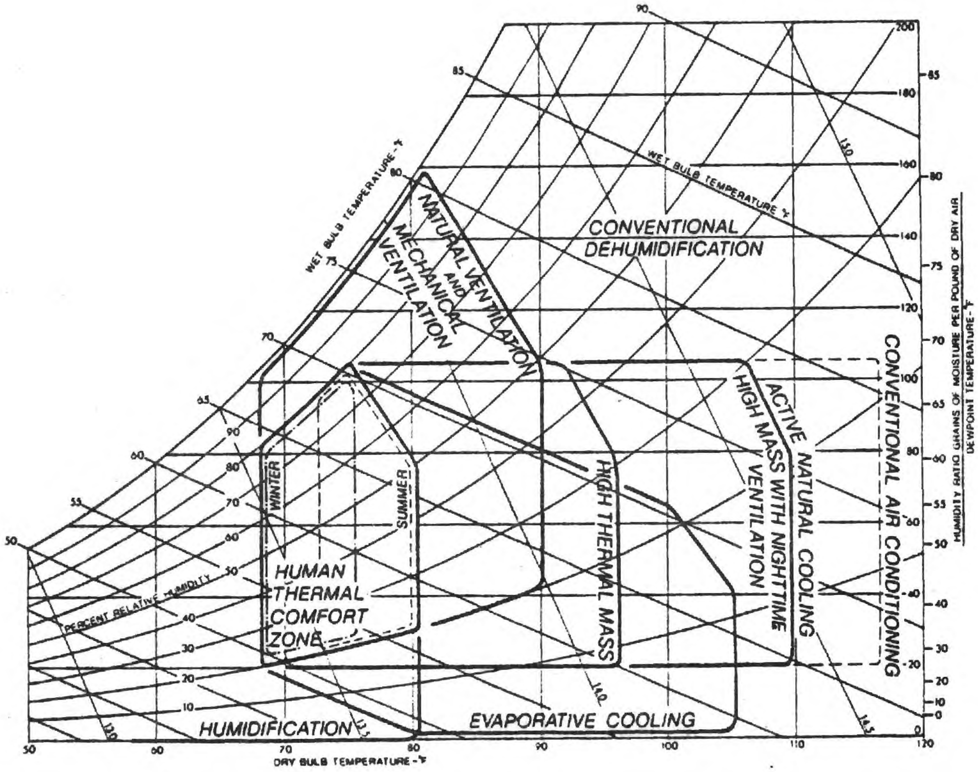

Figure 2.1 Thermal Comfort - Methods

of a solar chimney. Since wind velocities are almost always above these values even in the most calm areas of the United States[4], one must question this use of solar energy. A far more serious problem with thermally induced ventilation results from solar energy being available primarily during those times when the outside temperatures are the highest. This means that a solar chimney works best during those times when one would prefer to keep the heat outside the structure. It is difficult to visualize the solar chimney as a viable passive cooling system in any climate except during spring and fall when temperatures and relative humidities are moderate. Most of the thermal chimnies built in the Southeast have also increased the thermal load on the residence. When the chimney is incorporated as an integral part of the building, it has proven difficult to prevent some thermal migration from the chimney to the building living spaces.

2.3 Wind Scoops

Wind scoops are either fixed or moveable devices usually located on the roof or on specially built chimneys. These devices are designed to face into the prevailing wind and funnel it into the structure. Wind scoops are not new, having been used in Asia for centuries. Recent research shows that scoops which are moveable are much more effective than stationary scoops. Due to their limited area, it is doubtful that even moveable scoops are as effective as a continuous ridge vent which can be coupled to the living space.

As with all ventilation devices, wind scoops are effective during those periods of the year when temperatures and relative humidities are moderate. This limits their use to spring and fall in the southeast or to night-time, as will be discussed in the following section.

2.4 Night-Time Ventilation

This had proved to be the most effective cooling technique for the southeast before the advent of mechanically driven air conditioning, and is still used by older residents in many older homes. Basically, the structure is opened (highly ventilated) at night when the ambient temperatures are low. Ventilation may be natural, with window fans or with attic fans. The mass of the structure is cooled with the lower temperature night air. When the temperatures begin to rise the next morning, the structure is closed up and remains closed until the following night.

This passive cooling technique is effective in much of the southeast although the temperature and relative humidity sometimes fall outside the comfort region. Use of ceiling fans or similiar devices which will move the air about within the structure are frequently sufficient to result in comfortable conditions.

Tests by the author over the past three summers in Atlanta indicate that conditions can be kept quite comfortable much of the summer in a house designed for artificial air conditioning throught the use of night ventilation. These tests did reveal that problems may arise from carpets and furniture acting as dessicants, resulting in occasional mildew problems.

Use of night-time ventilation supplemented with mechanical air conditioning late in the day has been observed in a number of homes in the southeast. At first glance this appeared to be of questionable value due to the air conditioner having to remove the moisture introduced into the house during the night. The technique was shown to have considerable merit when one realized that when the house is closed in the morning only one house volume of air was trapped. Since most houses, even those being built today, have at least one volume of air infiltration per hour, the night-time ventilation did not significantly increase the latent load and did significantly decrease the sensible load. The significant increase in population and the resultant increase in crime rate has led many former followers of natural ventilation to abandon the concept because of the difficulty in maintaining residential security. Windows or doors through which air can pass usually are easy for an intruder to penetrate.

Despite its past use and sometimes present use in the southeast, ventilation would appear to not be a promising passive cooling technique for hot-humid climates, especially when daytime temperatures get above 32.2°F (90°F) and nighttime temperatures remain above 26.7°C (80°F). The author lived in a non air-conditioned house in South Georgia for the first 18 years of his life. The house used nighttime ventilation and an inside circulating fan to <u>decrease discomfort (not attain comfort)</u>. Memories are still vivid of the difficulty in sleeping during the hot summer nights. Despite air movement through the house and over the occupants, the side of the occupant next to the bed receives no benefit from this air movement. One quickly reaches a state where perspiration has soaked the bed and discomfort increases further.

3.0 EVAPORATIVE COOLING

Evaporative cooling is a time-proven concept and is particularly effective for arid climates. It works on the principle that as water is evaporated into air, the heat of vaporization of the water decreases the temperature of the air. As an example, if .002 Kg (.0047 lbs.) of water is evaporated into .45 Kg (1 lb.) of air at 32.2°C (90°F) with a relative humidity of 20% the air after evaporative cooling would be 20.6°C (69°F) and have a relative humidity of 70%. Unfortunately, the sourtheast is hot and humid during the summer months. The air already contains more moisture than is comfortable, thus further evaporation would not result in increased comfort.

While evaporative cooling usually cannot be used as effectively in the southeast for ventilation air, it can be an effective means of reducing sensible loads through roofs. Bacon[5] and Sutton[6] each conducted experiments showing that roof temperatures could be reduced by as much as 33.3°C (60°F) through the use of roof sprays. Since roof thermal loads may be a significant part of a residential sensible load, roof sprays are effective (especially for dark colored roofs), providing water is readily available.

4.0 RADIANT COOLING

Radiant cooling works on the concept that night sky temperatures are much lower than are surface temperatures near the ground. If a large surface is faced toward the sky, there may be a significant temperature difference potential between the surface and a clear night sky. If the long wavelength emissivity of the radiating surface is high, the surface will radiate energy to the sky resulting in a decrease in surface temperature.

Experiments on a radiant cooling systems called "Skytherm"[7] at Atascadero[8], California showed this concept to have considerable potential in that climate, although evaporative cooling was used to augment its performance. Computer simulations of "Skytherm" type systems at Trinity University[9] have shown the concept to have potential in much of the United States.

Radiant cooling becomes much less effective as atmospheric moisture levels increase, as atmospheric pollution increases, and as cloud cover increases. Atmospheric moisture levels are very high during the summer months in the southeast. While radiant cooling systems may still have some potential for meeting part of the sensible cooling load, they are totally incapable of meeting the latent loads experienced in the southeast. There are also considerable questions concerning market acceptance of the architectural restraints resulting from the "Skytherm" concept or even the alternatives proposed by Thomston[10] and Givoni[11].

McCathren[12] has recently conducted experiments to determine the radiative cooling potential in Atlanta. He used radiators much like active collectors and determined that while some radiative cooling potential did exist, this potential was so low that there is a serious question of economic viability.

5.0 EARTH TEMPERING

Earth tempering is the term presently used for structures which are buried, or semi-buried and use the ground both as insulation and as a thermal mass to moderate temperature differentials across building elements. When properly designed, these buildings usually perform quite well unless the building has a high internal heat gain such as might be the case with industrial or large commercial buildings.

The degree of success is usually highly dependent upon the depth at which the building is placed, as well as the quantity, size and the orientation of openings to the surface. At most latitudes in the United States, it is desirable to have some glazing facing south either directly or through an atrium.

The most difficult problem facing earth tempered buildings in hot-humid climates is the same problem faced by all passive cooling techniques, humidity control. One does not want earth tempered buildings to carry any latent load because it would manifest itself as water condensation on walls, floors, or ceilings. This is undesirable from comfort, health and aesthetic standpoints. Latent load control is discussed later.

6.0 ICE HOUSE ROOFS

An ice house roof is essentially a second roof built above and spaced .20-.36m (8-14 in.) from the conventional roof. In its conventional form, it is open at the bottom and at the roof peak. Thus energy absorbed by the upper roof induces convective flow between the two roofs. This flow carries away most of the radiant energy falling on the roof before it can enter the conventional roof.

This may be a viable cooling concept although little work has been conducted to quantify its performance. Morgan[13] and Edgerly[14] have conducted limited tests indicating good performance. Further work is necessary to develop analytical methods simple enough for every day use. The conventional ice house roof should be classified as a load reduction technique rather than a passive cooling technique.

A modification of this concept should improve performance. The upper surface should be white or very light colored. The underside of the top surface should be highly reflective. This minimizes the energy absorbed by the upper roof and also minimizes the energy radiated to the lower roof. If the inlet and discharge vents of the ice house roof can be closed at night and air from within the structure circulated between the two roofs, the upper roof can be used as a nocturnal radiator. It would provide significant cooling on clear nights at less cost than most nocturnal radiation techniques while also reducing loads during the day.

7.0 VARIABLE INSULATION

Givoni[15] and others have demonstrated that use of insulation with light colored roofs and walls greatly reduced internal temperatures during the day, but resulted in higher average internal temperatures at night. This suggests that a structure with a variable insulation capability could be used to depress both day and night internal temperatures. There are a number of approaches to variable insulation although most are quite complex. This passive cooling technique needs considerable experimental work before it can be listed as a viable passive cooling technique. It does have some promise. One would expect performance similar to nighttime ventilation without the increased humidity problem which frequently exists in hot-humid climates with direct ventilation.

8.0 COLOR

Color must be considered one of the cheapest and most effective load reduction techniques available. Whites or other light colors reflect 85-95% of the solar energy because of very low absorptivities at short wavelengths. These colors have very high (.9 or greater) emissivities at long wavelengths and are very good radiators. Thus, they not only reflect most of the sun's thermal energy, they also radiate energy to the sky and other cooler objects. Givoni[1] has shown significant reductions in internal gains and temperatures through the use of white-washed walls and roofs.

9.0 EARTH TUBES

These are simple large .15-.76m (6-30 in.) diameter pipes of metal, plastic, concrete, etc., buried in the ground .91-2.4m (3-8 Ft.) below the surface. Ventilation air for the building is pulled in from an open end exposed above ground. The open end is covered with a screen to prevent rodents, insects, etc. from entering the tube. The object of burying the tube is to take advantage of the relatively constant and moderate temperatures at depth below 6-8 feet. The temperature profile below the surface lags ambient temperature profile by days, weeks and months, depending upon depth. The lag increases with depth; in mid summer, the ground temperature is approximately 16.5°C (62°F) at 7.6m-9.1m (25-30 Ft.) but increases to 23.7-26.4°C (75-80°F) at 1.8m (6 Ft.) in the Atlanta area. Air pulled through this tube cools off. If the tube is long enough and the air flow is not great, the air can approach the ground temperature. Energy from the air increases the ground temperature adjacent to the tube thus decreasing cooling potential.

Several computer simulations[16-17], indicate that ground conductivity is not important and that heat transfer is limited by the heat transfer coefficient between the air and tube. A recent study by Shelton[18], however, indicates that ground conductivity may be important for earth tubes.

Other problems exist. Due to the high relative humidity in the southeast, the tubes, unless very long, only cool the air to the dewpoint temperature and thus do not remove any of the latent load. The air being discharged into the residence is nearly saturated, i.e., has a 100% relative humidity, and can be quite uncomfortable. If condensation does occur in the tube as would be the case on very humid days or nights in early summer, the water must be properly drained or conditions ideal for mold, mildew, bacteria, etc., will exist in the tube, possibly creating a health problem. As mentioned earlier, these conditions are similar to those found to be growth centers for Legionairs disease.

There appear to be two modifications to the concept which show promise. If a misting chamber (evaporative cooler) is put on the inlet side the air temperature can be dropped to the wet bulb temperature and the earth tube used to remove and further cool the air through condenstation. One still has air at 100% relative humidity at discharge which likely will be uncomfortable. The second modification is to use the air from the earth tube to cool the mass in the building rather than as ventilation air. This will not decrease the latent load, but it also will not add to it as it might when used as ventilation air. It will reduce the sensible load. Because of the high interest in earth cooling tubes, and because of experimental data, as well as a good theoretical model, becoming available due to another research program with which the author was involved, they will be discussed extensively in Chapter III.

10.0 SORBENT DEHUMIDIFICATION

It has been suggested in several popular passive papers that since high humidity is the most difficult problem in the southeast, one could use a sorbent (either adsorbent or absorbent) to remove moisture, thus removing the humidity problem. Unfortunately, when one removes moisture through the use of sorbents, the heat of vaporization and, depending upon the sorbent, possibly the heat of absorption as well, will be released. This raises air temperature significantly. Figure 2.2, taken from ASHRAE Equipment Handbook[19] shows the performance of a typical sorbent dehumidifier. This figure shows that removing 3.37gm(.0074 lb.) of moisture per .45Kg (1 lb.) of dry air will result in a 2.2°C (36°F) increase in air temperature. Sorbent dehumidification is almost exactly the opposite of evaporative cooling, thus one would expect the air to follow closely the constanct enthalpy line on the psycholmetric chart, as shown in Figure 2.3.

Various passive sorbent dehumidification techniques have been proposed[20-21]. When one completes the design for these systems including all of the equipment necessary to make them perform satisfactorily, one realizes that the system closely resembles active sorbent cooling systems such as the Solar-Mec System[22] which has been under investigation for years.

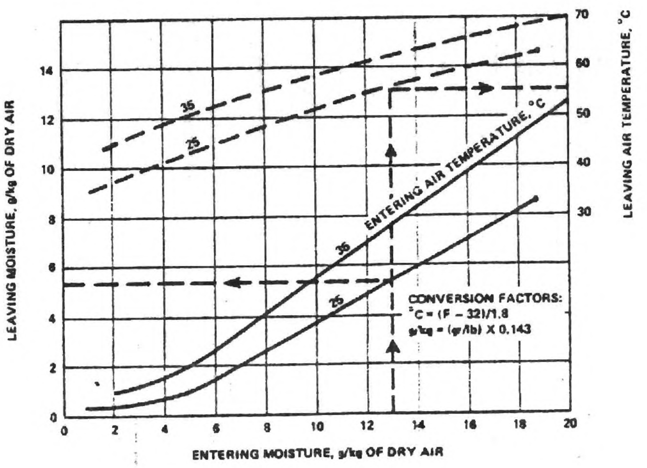

Figure 2.2 Air Temperature Increase with Sorbent Dehumidification

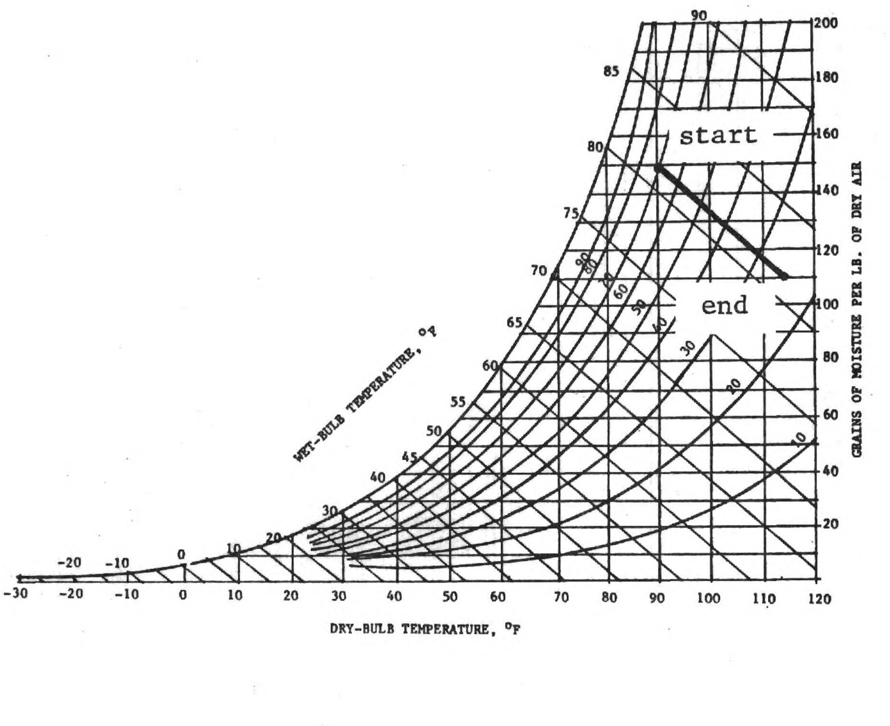

Figure 2.3 Sorbent Dehumidification on Psychrometric Chart

11.0 DETACHED EARTH TEMPERING

One of the earth tube modifications discussed as showing promise was the use of the earth tempered air to cool the structural mass rather than as ventilation air. This has led to a passive cooling concept which we have called "Detached Earth Tempering" (DET). This concept promises the advantages of below ground earth tempered structures without the structural, moisture, site and cost problems usually associated with below grade buildings. It also permits control of the degree of tempering to meet varying ambient conditions. This is not possible with below grade structures and is one of the greatest problems presently encountered with earth tempered structures. Designers presently have inadequate tools to determine where and how much insulation should be added to below grade structures. If the wrong location or thickness is chosen, little can be done after the structure is constructed. The Detached Earth Tempering concept promises the capability of complete control.

If the fluid cooled in the tube is used to cool the structure rather than being introduced into the occupied space, air might not be the best fluid. Glennie[16] and Sheldon[18] have shown that multiples of smaller tubes work better than one large tube. Large diameter pipes of any type may be prohibitively expensive for earth tube application due to pipe cost and installation costs. If the earth's capacity to cool is to be used, and the air-to-pipe heat transfer coefficient is the limiting factor, it could be that a much longer and smaller closed loop buried pipe using water as the transfer media would be cheaper to purchase and install, and would perform much better. Pumping power would be less because water is much easier to pump than air. The water to tube heat transfer coefficient would be sufficiently high that ground conductivity would definitely be the limiting factor. Three methods of detached earth tempering appear feasible. Each has advantages and disadvantages. If ambient or internal latent loads are low one could use earth cooled air to ventilate. The water could be pumped through a relatively small finned tube heat exchanger over which the air could be pulled. A two foot diameter pipe 30.5m (100 Ft.) long only has 58.34m (628 ft^2) of heat transfer area. A finned tube heat exchanger with two to four times this transfer area would be only .91m x.91m (3 Ft.x 3 Ft.) and would cost considerably less than the cost of the two foot diameter tube. The misting chamber could still be used and the condensate problem becomes one easily handled as with conventional air conditioners.

One is still faced with air discharge at a high relative humidity if one uses the cooling capacity of the earth to cool ventilation air. A second approach could be to bury (imbed) the water pipes in the structure of the house thus cooling the mass and removing the sensible load. Latent loads would be reduced and controlled as discussed later. If the mass of the house is inside the insulation, the water cooled structure becomes essentially a buried house in which one can control the degree of earth tempering by varying the water flow rate. This passive cooling technique will work equally well as a passive heating system because the ground temperature (in winter) would provide a stable relative high temperature on the structure.

A third approach would be to not use the water to cool the structure directly but to use a water to air heat exchanger to provide earth tempered air which would then temper the structure. The use of air rather than water may be easier to accomplish from a construction standpoint.

Several other sources of earth tempered water are available and should be thoroughly investigated. In areas with high water tables, a shallow well will provide water at a constant temperature. In areas where water availability is low, a vertical earth to water heat exchanger may be easier to install and may perform as well or better than the horizontal exchanger.

There are many unknowns in all passive cooling concepts. The DET concept shows promise and perhaps has fewer unknowns than other concepts. This concept was identified as the one which should be most vigorously investigated on this research program. Chapters IV of this report will detail the experimental and theoretical progress made toward developing this concept as a viable passive cooling technique for hot-humid climates.

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CHAPTER III

EARTH COOLING TUBES

Earth cooling tubes are popularly touted as a viable method of passive or low-energy space cooling. Despite a lack of basic research, operating experience, and design guidelines, the concept has been well received, resulting in an ever increasing number of new installations.

Until recently there was not a good thermal model with an easily understood approach to understanding passive cooling with cooling tubes. Most installations were made with the attitude of lets try it and see how well it works. More often than not, it worked very poorly. This chapter presents a performance model of earth cooling tubes in an attempt to arrive at fundamental conclusions about their cooling potential. The applicability and limitations of the model are discussed as are an ongoing field monitoring program and model verification efforts. The results of sensitivity studies on performance variables facilitate consideration of design variations. It is hoped that this discussion will cause more care to be exercised in the design of earth cooling tubes and more restraint to be exercised in their use.

1.0 INTRODUCTION

Four heat sinks are available for use in space cooling processes: the atmosphere, water (surface and subsurface), deep space, and the earth. In many areas, notably the southeastern United States, climatic conditions limit the rejection of heat to those sinks involving radiation, evaporation and diurnal temperature variations.

In such cases, the earth may offer the only sink available at temperatures amenable to "passive" or low-energy space cooling. Architectural designs which incorporate direct building/earth contact and earth cooling tubes which temper air by pulling it through buried tubes, provide two methods of utilizing the earth sink. Earth cooling tubes minimize the impact on the building design and offer retrofit possibilities. Furthermore, they allow the building to be alternatively thermally coupled to the earth or isolated from it in response to the building's cooling requirements.

Offering the lure of unsophisticated hardware, low operating energy and theoretical simplicity, the earth cooling tube concept has been eagerly embraced by the energy-conscious popular media and public. Earth cooling is now viewed by many, perhaps incorrectly, as an economical means of passive cooling and dehumidification for all climates.

2.0 THERMAL MODEL

Although a large number of cooling tubes have been installed, few, if any, have been analyzed to determine their performance or optimize their design. The attitude seems to be to install something and take what cooling results, if any. This has resulted in many thousands of dollars being wasted on designs which have not been optimized and which perform very poorly, leading many people to become dissatisfied with cooling tubes.

Until recently, most cooling tube simulation routines were either too simple to be of benefit or too complicated for practical application in design. Historically, accurate simulation routines were based on finite difference nodal solutions which require large expensive computers. Recently, Abrams, Akridge and Benton[1] developed a simplification of a line source equation developed by Ingersoll[2] which can be used with hand held programmable calculators to predict the performance of cooling tubes.

Ingersoll showed that energy transfer from a source can be defined by the following equation:

$$(T_{ts} - T_e) = \frac{Q'*r^{(2-n)}}{2\pi^{(n/2)}k} \int_{r\eta}^{\infty} \beta^{(n-3)} e^{-\beta^2} d\beta \qquad (1)$$

where:

T_{ts}	=	outside tube surface temperature (^oF)
T_e	=	far field earth temperature (undisturbed soil temperature) (^oF)
Q'	=	rate of heat flow per unit of tube length (Btu/hr ·ft)
r	=	tube radius (ft)
n	=	number of dimension of flow (n=2 for a line source)
k	=	thermal conductivity of soil (Btu/hr·^{o}F·ft)
η	=	$\frac{1}{2\sqrt{t\alpha}}$ (1/FT)
α	=	thermal diffusivity of soil (ft^2/hr) = $K/c\rho$
β	=	variable of integration
c	=	specific heat of soil (Btu/lb^{o}F)
ρ	=	density of soil (lb/ft^3)
t	=	time after start (hrs)

For a line source, equation (1) reduces to:

$$(T_{ts} - T_e) = \frac{Q'}{2\pi k} \int_{r\eta}^{\infty} \frac{e^{-\beta^2}}{\beta} d\beta \qquad (2)$$

Equation (2) treats the energy transfer as a line source and assumes negligible heat transfer along the axis of the tube. Values of the integral may be found in table of integrals, or for values of $r\eta$ less than 0.6, the integral may be represented by:

$$\int_{r\eta}^{\infty} \frac{e^{-\beta^2}}{\beta} d\beta = \ell n \frac{1}{r\eta} + \frac{(r\eta)^2}{2} - \frac{(r\eta)^4}{8} - 0.2886 \qquad (3)$$

By substituting equation (3) into equation (2) one obtains:

$$(T_{ts} - T_e) = \frac{Q'}{2\pi k} * \left(\ell n \frac{1}{r\eta} + \frac{(r\eta)^2}{2} - \frac{(r\eta)^4}{8} - 0.2886\right) \tag{4}$$

While equation (4) is useful in calculating the change in temperature of soil adjacent to a tube as a function of energy withdrawal rate (Q'), time t, and soil properties; it is not in a form that is readily useful for calculating the performance of earth cooling tubes. If one solves for Q' and rearranges equation (4) into the form:

$$Q' = U_g * A_p * (T_{ts} - T_e) \tag{5}$$

where:

$$U_g = \frac{1}{R_g} = \frac{1}{\frac{r}{k} * \left(\ell n \frac{1}{r\eta} + \frac{(r\eta)^2}{2} - \frac{(r\eta)^4}{8} - 0.2886\right)}$$

A_p = area of pipe per ft. of length = $2\pi r * 1$

$(T_{ts} - T_e)$ = temperature difference between soil adjacent to pipe and far field soil temperature.

Equation (4) becomes:

$$Q' = \frac{2\pi k(T_{ts} - T_e)}{\left(\ell n \frac{1}{r\eta} + \frac{(r\eta)^2}{2} - \frac{(r\eta)^4}{8} - 0.2886\right)} \tag{6}$$

Since one is primarily concerned with the energy transferred to the fluid flowing through the pipe, one must also include the thermal resistance through the tube wall and the thermal resistance of the fluid film. Pipe wall resistance may be expressed as:

$$Rp = \frac{t_p}{k_p} \tag{7}$$

where:

t_p = pipe thickness in ft.

k_p = thermal conductivity of pipe $\frac{(Btu \cdot ft)}{hr \cdot ft^2 \cdot {}^oF}$

Pipe film resistance may be expressed as:

$$R_o = \frac{1}{h_o} \quad \frac{(Btu)}{hr \cdot ft^2 \cdot {}^oF} \tag{8}$$

where:

h_o = film transfer coefficient

Total thermal resistance to energy flow can now be expressed as:

$$R_t = R_g + R_p + R_o$$

$$R_t = \frac{r}{k} * \left(\ell n \frac{1}{r\eta} + \frac{(r\eta)^2}{2} - \frac{(r\eta)^4}{8} - 0.2886\right) + \frac{t_p}{k_p} + \frac{1}{h_o} \tag{9}$$

Energy transfer for a unit length of pipe can now be expressed in the form:

$$Q_x = A_p * \frac{1}{R_t} * (T_{fx} - T_e) \tag{10}$$

where:

T_{fx} = fluid temperature, ${}^oF$
T_e = far field earth temperature, ${}^oF$

This gives a final form of the expression:

$$Q_x = 2\pi r * \frac{1}{\left(\frac{r}{k} * \left(\ell n \frac{1}{r\eta} + \frac{(r\eta)^2}{2} - \frac{(r\eta)^4}{8} - 0.2886\right) + \frac{t_p}{k_p} + \frac{1}{h_o}\right)} * (T_{fx} - T_e) \tag{11}$$

Equation (11) allows one to calculate the energy transfer per foot of pipe length based on far field soil temperatures (T_e) and fluid temperatures (T_{fx}) at position x along the length of pipe. Obviously as energy is lost or gained at a position x, the fluid exit temperature T_{fx+1} will have changed. This change can be calculated by:

$$T_{fx+1} = T_{fx} + \frac{Q_x}{\dot{m}C_p} \tag{12}$$

where:

T_{fx+1} = fluid temperature at position x+1
T_{fx} = fluid temperature at position x
Q_x = energy lost or gained between positions x and x+1
$\dot{m}$ = fluid mass flow rate (lbs/hr)
C_p = fluid specific heat (Btu/lb°F)

Total energy lost or gained along the length of pipe of length L is:

$$Q_T = \sum_{0}^{L} Q_x = \dot{m}\, C_p (T_L - T_o) \tag{13}$$

where:

Q_T is total energy transfered (Btu/hr)
T_L is fluid temperature at x = L
T_o is fluid temperature at x = 0

3.0 MODEL LIMITATIONS

The equations developed here to estimate the performance of earth cooling tubes were developed using several simplifying assumptions. First, it is assumed that the soil is an infinite medium at a uniform temperature. At first, this would appear to be a very limiting assumption, since we know that the soil temperature not only changes with depth below the surface but also varies seasonally. The assumption does limit accuracy if the tube is at a shallow depth or is close to another tube. It is believed that the assumption is reasonably valid providing the tube is greater than six feet below the surface and is separated at least six feet from adjacent tubes. One should use the average monthly soil temperature for a depth comparable to the tube center line for the far field temperature. The best use of these equations lies not in quantifying performance but in parametric studies to determine the effect of different soil properties, pipe diameters, pipe materials and air flow velocities.

A second assumption is that the soil properties are uniform and constant for a particular simulation. In practice we know that these vary both with depth and time. Soil conductivity is highly sensitive to moisture content, becoming much greater at higher moisture contents. If one uses average soil properties for the area where the tubes are installed, this variation averages out over a year.

4.0 PARAMETRIC STUDIES

While it is believed that the equations developed here can provide reasonable estimates of cooling tube performance, their real value lies in parametric studies to determine the effect of tube diameter, tube length, mass flow rate, tube material and thickness, soil properties and time.

Figure 3.1 shows the temperature difference between the soil adjacent to the pipe and soil remote (undisturbed) from the pipe for a constant energy input of 50 Btu/Hr·Ft as a function of pipe diameter. Figure 3.2 shows the temperature differences if the energy per square foot of pipe is held constant. When one looks at Figure 3.1, one might conclude that larger pipe diameters are desirable. Figure 3.2 shows this not necessarily to be the case. A two-inch diameter pipe, two feet long, has much less temperature differential than a four-inch diameter pipe, one foot long, for the same total energy transferred. This becomes even clearer in Figure 3.3 where

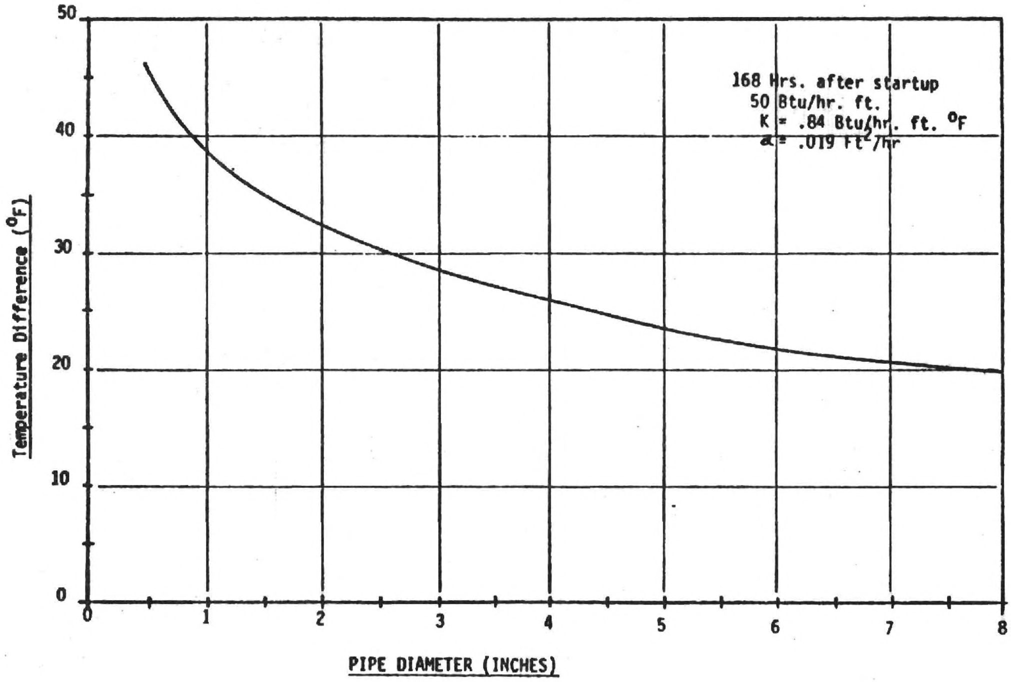


Figure 3.1 Temperature Difference as a Function of Pipe Diameter

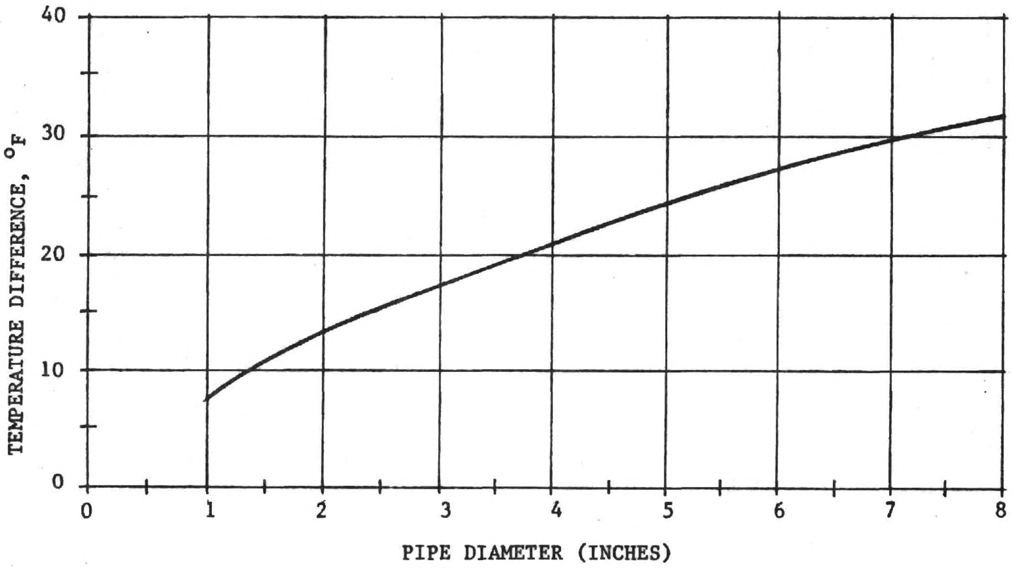


Figure 3.2 Temperature Difference for Constant Energy Input

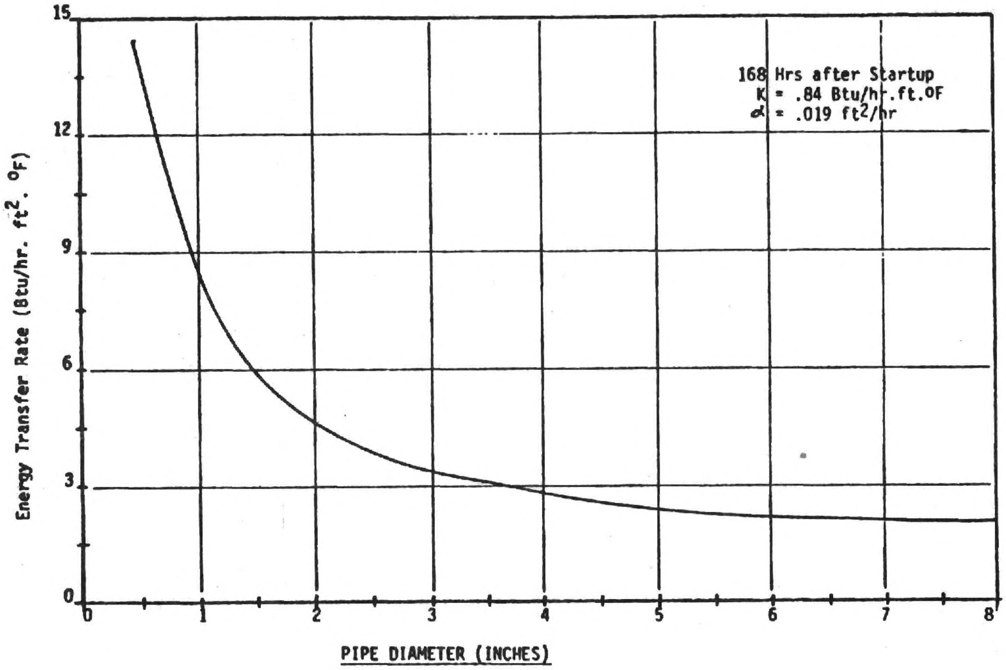


Figure 3.3 Energy Transfer Rate as a Function of Pipe Diameter

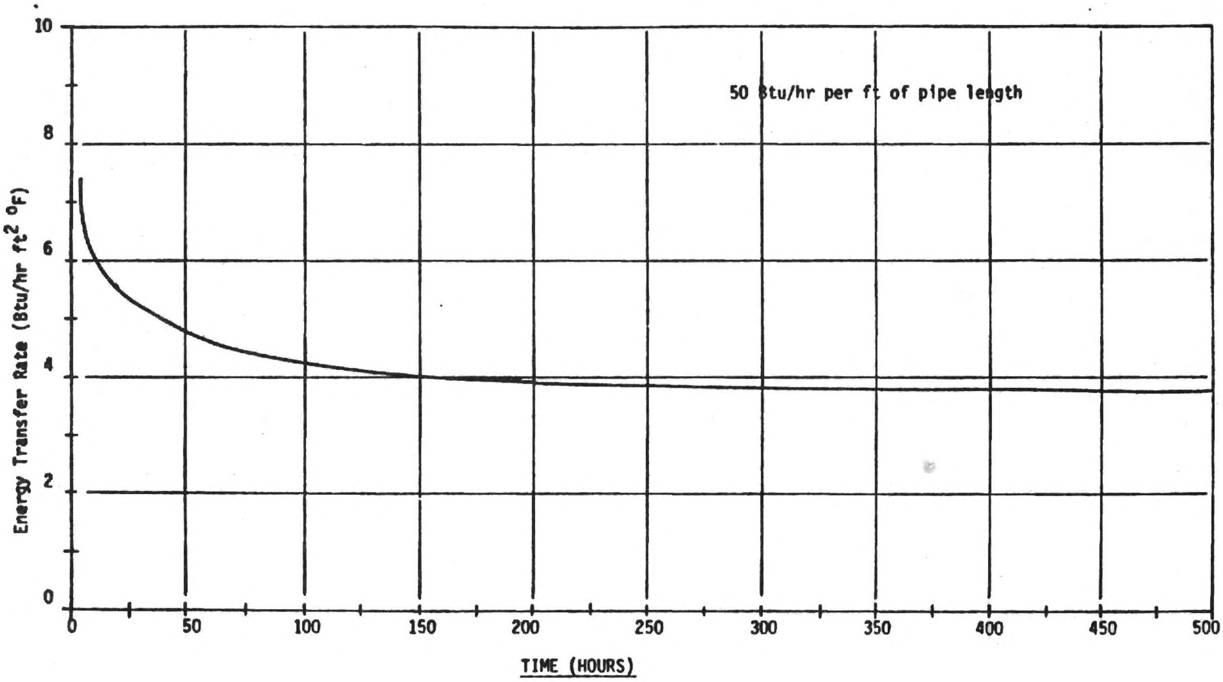


Figure 3.4 Energy Transfer Rate as a Function of Time

energy transfer rate (Btu/hr.ft^2.^{o}F) is plotted versus pipe diameter. Figure 3.3 shows that smaller pipes have much higher transfer rates for a given wall area.

Basically, Figures 3.2 and 3.3 show that smaller, long pipes are better than larger, short pipes. This is not unpredictable if one recognizes that energy flow to the pipe is essentially radial and that small, long pipes are exposed to more undisturbed soil than large, short pipes.

While the calculations show that the smaller the pipe, the better its operation from an energy transfer standpoint, there are other factors, such as pressure drop through the pipe, trenching costs, and available space, which will influence pipe diameter selection.

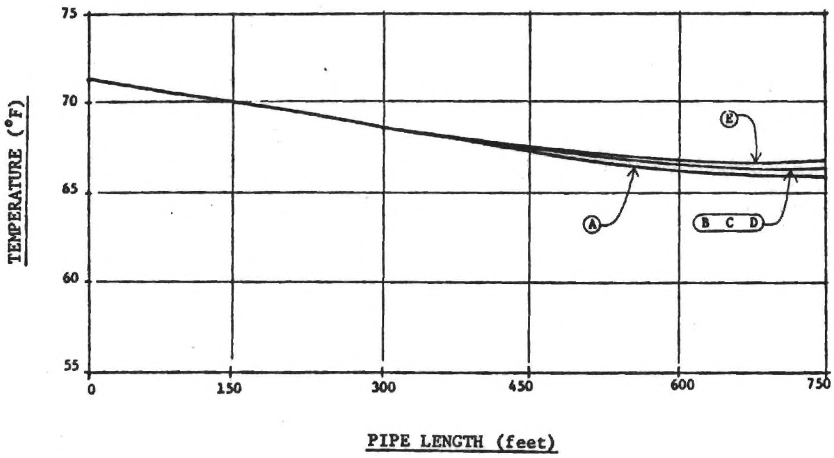
Figure 3.4 shows the effect of time on energy transfer rate. This figure shows energy transfer rates to drop quite rapidly during the first 50 hours (2 days), drop less rapidly during the next 100 hours (4 days), and essentially become constant after 168 hours (7 days). For a tube operating continously one could estimate cooling rate by using a time greater than 168 hrs.

Figure 3.5 shows the effect of soil diffusivity on temperature difference. Comparison of Figure 3.1 and 3.5 shows the lower thermal conductivity and diffusivity significantly increase the temperature difference required for a given heat transfer rate. It is believed the values used to calculate the data used to construct Figure 3.1 are more nearly representative of the soil in Atlanta.

Figure 3.6 shows the effect of mass flow rate on exit temperature. As one would expect, higher mass flow rates result in higher exit temperatures. Higher flow rates also result in higher energy transfer rates.

A number of conclusions may be drawn from these parametric studies:

1. Small diameter tubes are more effective per ft^2 of surface area than are



SOIL DIFFUSIVITY		HEAT TRANSFER AT END OF RUN
CURVE A =	.02 SF/HR	TOTAL Q = 11,625 BTU/HR
" " B =	.025 SF/HR	" " " = 11,403 BTU/HR
" " C =	.03 SF/HR	" " " = 11,228 BTU/HR
" " D =	.035 SF/HR	" " " = 11,084 BTU/HR
" " E =	.04 SF/HR	" " " = 10,962 BTU/HR

Figure 3.5 Effect of Diffusivity

MASS FLOW RATES		HEAT TRANSFER AT END OF RUN
CURVE A =	100 LBS/HR	TOTAL Q = 1,699 BTU/HR
" " B =	500 LBS/HR	" " " = 6,766 BTU/HR
" " C =	1000 LBS/HR	" " " = 9,319 BTU/HR
" " D =	2000 LBS/HR	" " " = 11,144 BTU/HR
" " E =	5000 LBS/HR	" " " = 12,483 BTU/HR

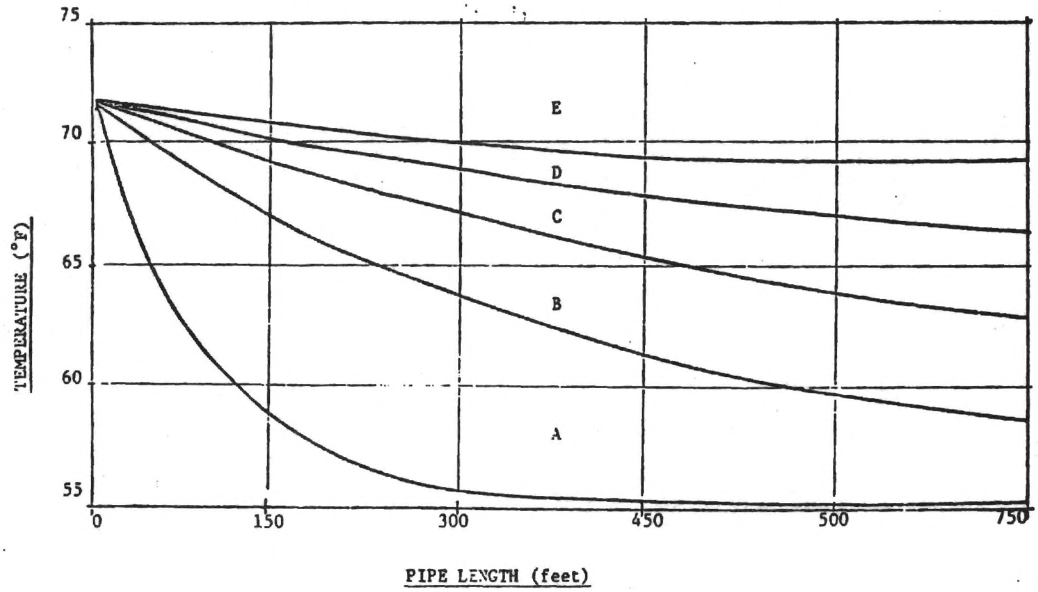


Figure 3.6 Effect of Mass Flow Rate on Exit Temperature

larger tubes.

2. Air reaches the adjacent ground temperature relatively quickly so long tubes are unnecessary.

3. A large diameter tube of a unit length will transfer more energy than a small diameter tube of unit length, thus multiple small tubes should be used.

4. Tubes should be placed as deeply as possible so that the far field temperature is the lowest and relatively constant.

5. Ground temperatures and energy transfer rate during the first 168 hours of operation result in soil temperature adjacent to the tube which are too high in Atlanta for dehumidification.

6. Exit temperature is highly dependent upon the inlet temperature, thus use of open loop earth cooling tubes results in higher exit temperatures, higher adjacent soil temperatures, and high relative humidity.

7. Closed loop cooling tubes where the air is pulled from the house through the tube and back into the house are more effective than open loop systems because of the factors discussed in 6 above.

8. Ground thermal resistance is so high that tube thermal resistance is immaterial, i.e., plastic or concrete tubes perform as well thermally as copper tubes.

9. While higher mass flow rates result in greater cooling, this results from more air being cooled rather than higher film coefficients.

All of the discussion of earth cooling tubes thus far has centered around their operation in a continuous mode, i.e., they are turned on and stay on until the end of the summer. In practice, this is not how they are operated. They usually are turned on in the late morning and turned off in early to mid evening. This means that energy is not being dumped into the ground continuously. There is a soak period when the adjacent soil temperatures are permitted to adjust toward the far field temperatures.

Figure 3.7 shows measured data taken on a 21" diameter, 100' long aluminum earth cooling tube in Atlanta. Notice that the temperatures adjacent to the tube rises during the time the tube is operating and then drops when it is off. An estimate of the performance of a tube operating in this manner can be developed by ratioing the on time to the total time. If the tube is only operated 1/3 of the time, the tube will take three times as long to reach a stable operating condition as it would if run continuously.

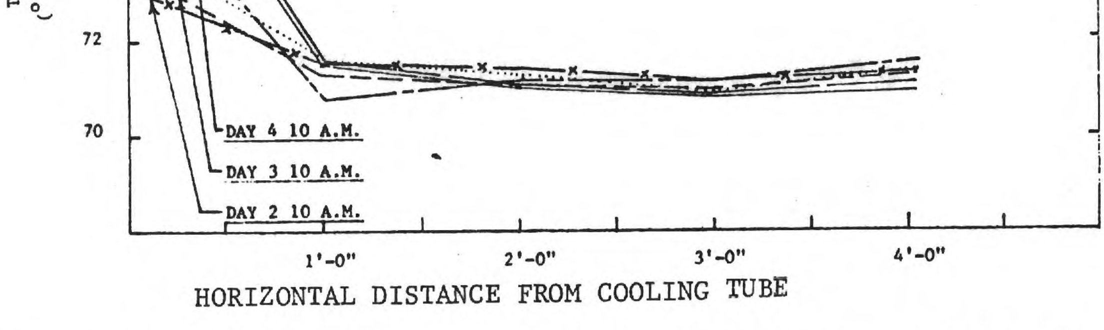


Figure 3.7 Adjacent Ground Temperature With Intermittent Operation

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CHAPTER IV

PASSIVE COOLING FOR HOT HUMID CLIMATES

BASIC CONCEPT

This study did show that earth tempering, in the form of underground construction, could significantly reduce sensible cooling loads in hot-humid climates if air infiltration could be controlled. Despite this potential, few examples of earth tempering in hot-humid climates exist. Unlike most passive techniques, earth tempering in hot-humid climates has been held back by a technical problem. Until recently, it was not possible to control infiltration sufficiently well to make earth tempering practical in hot-humid climates.

The importance of infiltration or ventilation control in hot-humid climates becomes very apparent when one looks at the latent and sensible loads of a building as a function of the ventilation (infiltration) rates while keeping the ventilation air temperature constant. If the relative humidity of the air is now varied, one finds the only difference in the thermal load on buildings in arid and humid climates is due to the latent loads caused by ventilation (infiltration). Obviously, if infiltration can be greatly reduced and carefully controlled, a building in a humid climate will perform nearly the same as in an arid climate.

Although the study showed heating and cooling potential for earth tempering, it also showed serious architectural and market constraints on conventional earth tempered (underground) buildings. The study also showed that many of the thermal advantages of underground construction may be realized above ground through the use of a concept we have chosen to call "Detached Earth Tempering" (DET). If architectural constraints prevent taking the building underground, the Detached Earth

Tempering Concept attempts to bring the thermal advantages of underground structures to above grade buildings.

1.0 BASIC CONCEPT

The basic concept behind Detached Earth Tempering is to bury coils in the earth through which water or other similar heat transfer fluids can circulate. The fluid having been cooled by earth contact can then be circulated through building elements such as the floor, ceiling or walls. If the walls are well insulated and the insulation is located on the outside of the structure, one will have a cool wall structure similar to that of an underground structure. If infiltration is controlled through the use of good seals, vapor barriers and air locks at the doors and ventilation is accomplished through the use of an enthalpy exchanger, the building will perform similar to an underground structure in an arid climate.

Initial computer studies showed that ground temperatures at depths of 1.2-3.6 meters (4-12 feet) are much too high in late summer to provide appreciable cooling. The computer studies were checked with ground temperature measurements. These measurements verified the predicted ground temperatures for areas well shaded. They also showed that areas with little ground cover can reach considerably higher temperatures.

Because high surface temperatures result in high temperatures at greater depths, one might minimize this effect by separating the surface from the soil at lower depths with insulation. Insulation of the soil from the surface also greatly reduces the rate at which energy can be lost to the ambient air during the winter months. This then requires that the soil beneath the insulation be cooled during the winter months if one is to have the low soil temperatures desired during the summer months.

Georgia Tech has installed an experimental field with 213 meters (700 feet) of 38 mm (1.5 inch) polyethylene pipe buried at a depth of 1.2 meters (4 feet) with .9 meters (3 feet) of dirt directly above, followed by 51 mm (2 inches) of extruded polystyrene insulation. The insulation is covered with .3 meters (1 foot) of dirt with a good sod cover. Ideally the field would be placed beneath the house to minimize undesirable ambient loads. Figure 4.1 shows a section of the Georgia Tech experimental field. Since it could not be installed beneath a house, great care has been exercised in providing a good sod cover to minimize radiant gains at the soil surface. Chapters V and VI discuss the computer simulation and experimental field in detail.

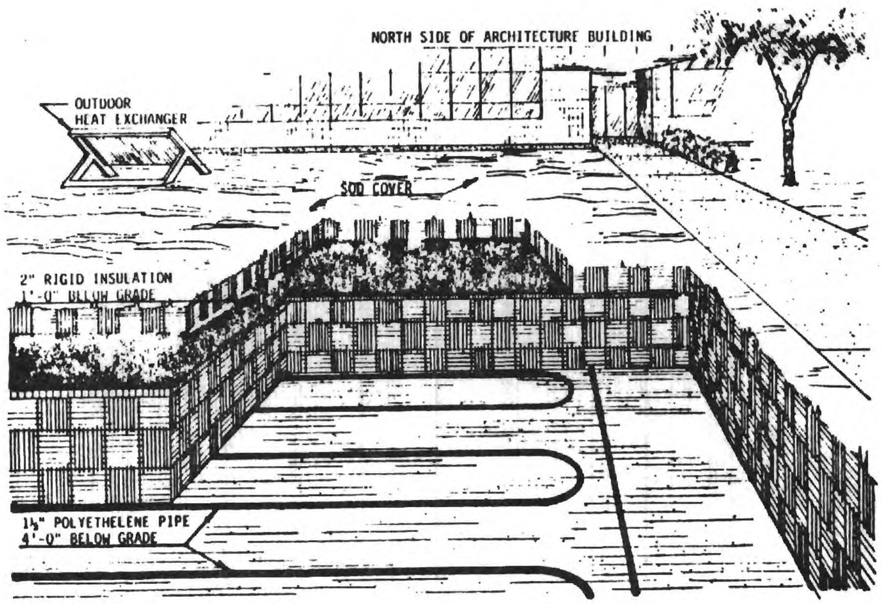


Figure 4.1 Exploded View Of Single Level Cooling Field

The insulated field is cooled during the winter months by circulating water through an above ground air-to-water heat exchanger and then through the buried coil. A differential thermostat turns on a small (1/15 hp) pump when the field is warmer than the ambient air.

The success of seasonal storage of cooling potential is highly dependent upon how one couples this cooling capacity to the occupants. Due to the low grade (temperature differences are relatively small) cooling potential, conventional cool air systems will not perform satisfactorily. Cooling through the use of building elements such as walls, floors or ceilings appears to offer the most potential. These elements provide large heat exchange surfaces in direct radiant contact with building occupants.

Initial computer studies showed that cooling capacity from a buried field might be incapable of meeting peak instantaneous loads but would be adequate for average daily loads. This indicated that optimum performance could not be obtained using low mass radiant planes. Although considerable data have been published in the literature directed toward the design of radiatively heated buildings, there are little data on design of radiatively cooled buildings. ASHRAE[1] provides some design guidelines using low mass radiatively cooling panels.

Radiative cooling potential of concrete walls of several thicknesses with several different tube spacings have been simulated using a thermal simulation program called MITAS[2] and a smaller thermal network program for microcomputers called T-NODE[3]. These simulations show the radiative cooling concept to have potential. The simulations have also shown the need for experimental data on the performance of such walls due to uncertainties about convective heat transfer coefficients on cooled walls.

A radiative panel test chamber has been designed, constructed and is currently being used to develop experimental data on the performance of cooling walls. Figure 4.2 is an exploded view of the radiative cooling test chamber. This chamber has the capability of quantifying the cooling performance of walls, floor and ceiling elements. Test results from the radiative cooling test chamber are discussed in a later chapter.

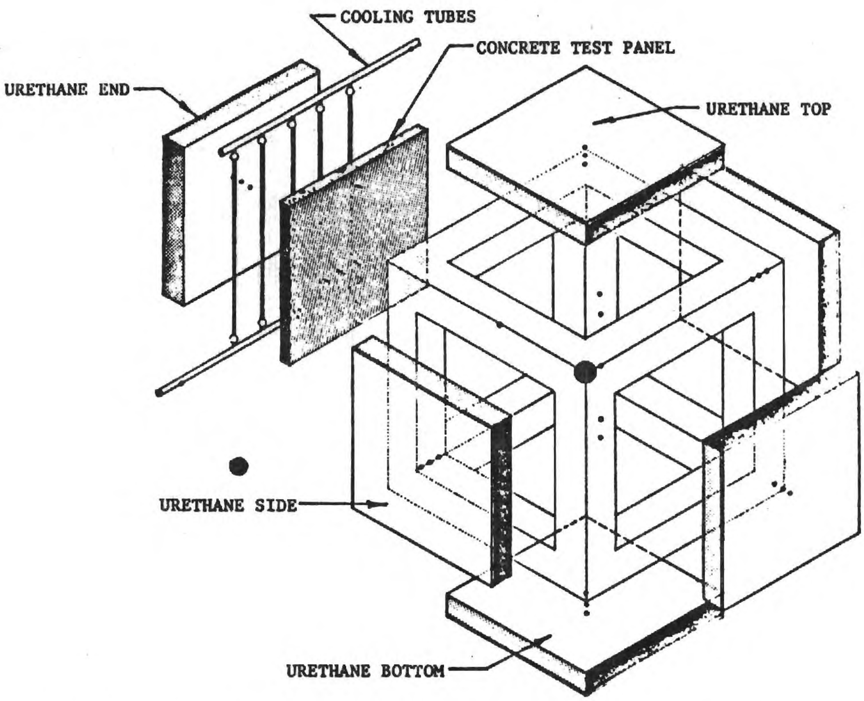


Figure 4.2 Exploded View of Radiant Cooling Test Box

2.0 AUXILLIARY SYSTEMS

It is imperative that passive cooling systems work well, or at least not interfere, with passive heating systems. It is also important that both passive heating and cooling systems be complemented with efficient auxilliary heating and cooling systems. Nothing useful is accomplished if much of the energy one saves with passive heating and/or cooling systems is lost through the use of inefficient auxilliary systems. Unfortunately many advocates of passive systems are opposed to incorporation of state-of-the art or high technology mechanical systems as a backup. This usually results in poor efficiency and less comfort.

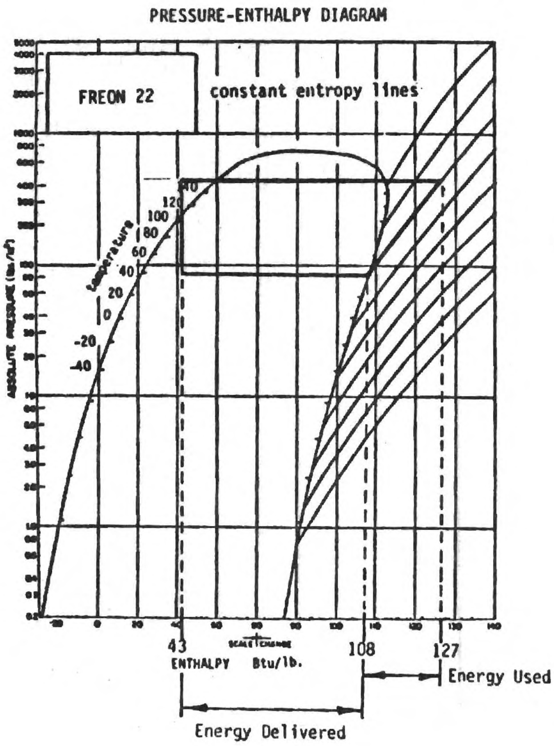
It was felt from the start of this program that it would be highly unlikely that a passive cooling system for hot-humid climates could be developed that would be capable of meeting 100% of a residential cooling load. This meant that an auxilliary cooling system was necessary if comfort was to be maintained. It was also felt that passive techniques for meeting latent cooling loads are not likely to be developed in the near future. If the sensible cooling load of a building is to be met radiatively in a humid climate it is imperative that latent loads be efficiently handled. This means that auxilliary systems are needed to handle latent loads at all times. They are also needed to handle both sensible and latent loads during extreme periods.

Unfortunately, if one were to employ a conventional air conditioner to handle the latent load, it would also provide sensible cooling which could be provided passively. It appears that greater efficiency can be obtained by handling the sensible and latent loads with separate equipment rather than with a single component as is normal practice.

2.1 Auxilliary Sensible Load

If one plots an idealized Rankine cycle air source air conditioner on a pressure enthalpy diagram one would have a cycle such as shown in Figure 4.3. The coefficient of performance (COP) would be about 3.42 for the most efficient systems presently available. One is limited to this COP by two factors. First, due to high ambient temperatures, one must have condenser temperatures of 150°F or above to dissipate the energy removed from the residence to the ambient air . One also must have evaporator temperatures at or below 50°F if one is to adequately handle the latent load.

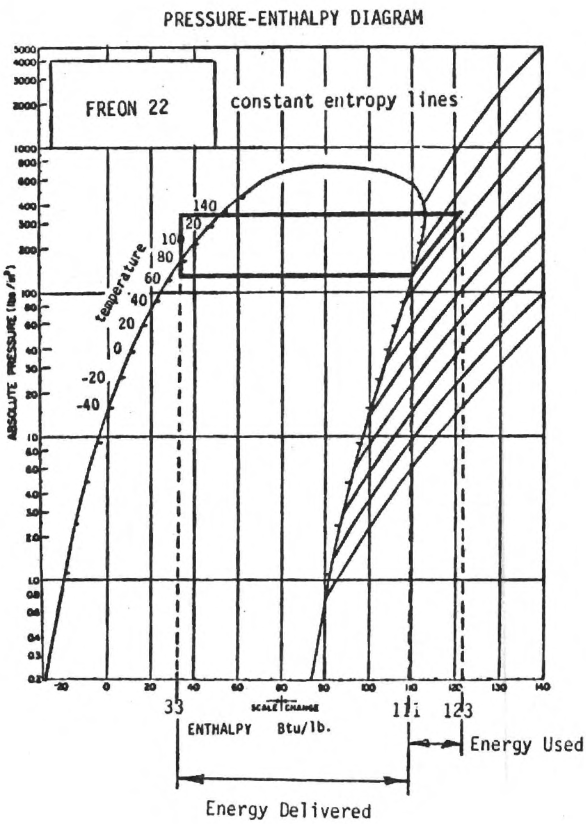
Use of a conventional air-source heat pump would not meet our stated desire to handle the sensible and latent loads separately. One can meet the sensible cooling load passively until the temperature of the water coming from the cooling field reaches 74-75°F. If one now supplies the 74-75°F water coming from the field to the condenser of a <u>water-to-water</u> heat pump and supplies water from the heat pump evaporator to the cooling walls, one can function with a Rankine cycle similar to the one shown in Figure



$$COP = \frac{\text{ENERGY DELIVERED}}{\text{ENERGY USED}}$$

$$COP = \frac{108-43}{127-108} = 3.42$$

Figure 4.3 Thermodynamic Cycle for Conventional Heat Pump



$$COP = \frac{\text{ENERGY DELIVERED}}{\text{ENERGY USED}}$$

$$COP = \frac{111-33}{124-111} = 6.0$$

Figure 4.4 Thermodynamic Cycle for Water-to - Water Heat Pump

4.4. Notice it is not now necessary to operate the condenser at 150°F because of the 74-75°F water available from the field. It is also not necessary to operate the evaporator at 50°F because the radiative cooling wall works well with 70°F water. One now has a auxilliary sensible cooling system with a COP of 6.0. Figure 4.5 gives a schematic of the cooling field and cooling wall when operating through the water-to-water heat pump.

Similar cycles can be shown for a conventional heat pump and a water-to-water heat pump in the heating mode. One finds the COP improves from 3.10 to 6.92 by going to a water-to-water heat pump. This system obviously meets our requirement for a high efficiency sensible auxilliary system which is compatible with the passive system.

2.2 Auxilliary Latent Load

If one succeeds in passively heating and cooling a residence 100%, one finds that a substantial energy requirement still remains. It is not uncommon for domestic hot water (DHW) energy requirements to exceed the heating and cooling requirements for well designed energy efficient conventional homes. Several manufacturers have recently marketed domestic hot water heaters which operate on a heat pump principle. These heat pump DHW heaters require only 40-50% as much energy input as conventional electric resistance DHW heaters. If one locates the heat pump DHW heater in occupied space it not only heats the domestic hot water more efficiently, it also provides sensible and latent cooling. Figure 4.6 shows a schematic for a heat pump DHW heater.

We now have an efficient DHW heater, a very efficient sensible auxilliary system and a latent auxilliary system which is a by-product of the DHW heater.

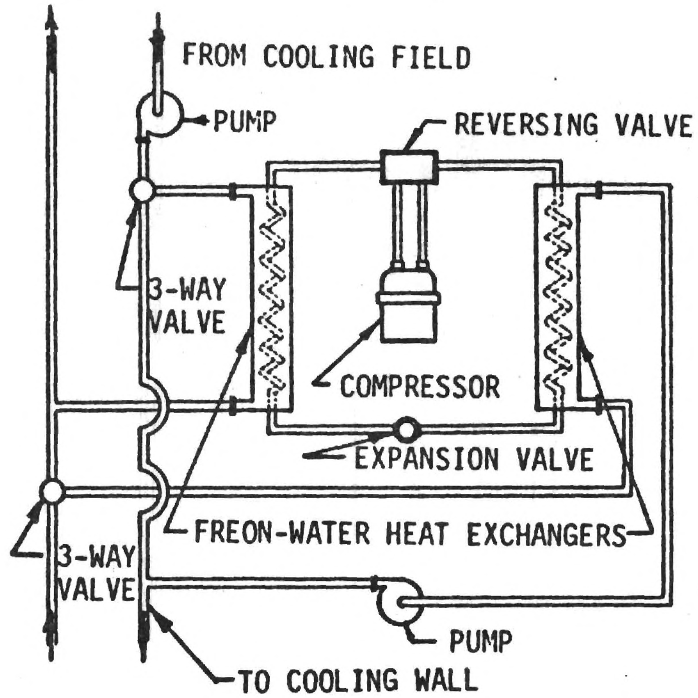


Figure 4.5 Water-to-Water Heat Pump Combined With Cooling Field

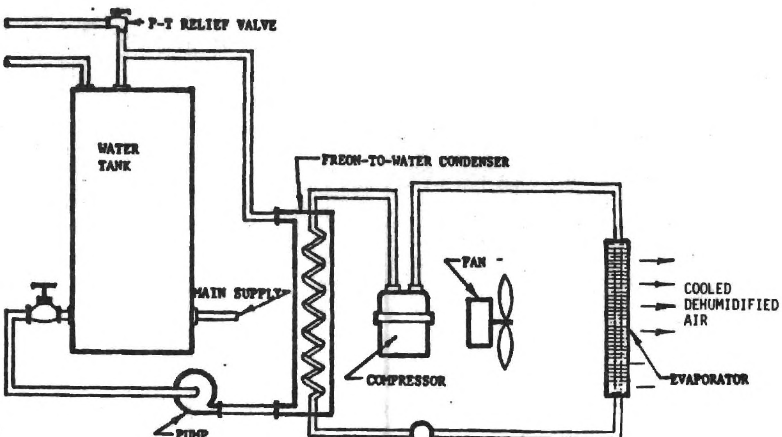


Figure 4.6 Heat Pump Domestic Hot Water Heater

3.0 ADVANCE MODE OF OPERATION

Once the auxilliary heating and cooling systems have been integrated into the passive design, one finds that a second and possibly better mode of operation becomes possible. One can passively cool with cooling potential stored in a block of earth until the water from the cooling field reaches approximately 23.1°C (74°F). When the water reaches 23.1°C (74°F) one actively cools with a water-to-water heat pump using the relatively cool 23.1°C (74°F) water from the cooling field. This increases the field temperature until it reaches perhaps 42.9°C (110°F) by the end of the summer. One can now passively heat using the 42.9°C (110°F) water coming from the field and the radiative cooling/heating wall. When the water coming from the field reaches approximately 29.2°C (85°F), the water is directed through the water-to-water heat pump and the heat pump used to heat through the radiative wall. This cools the field until at the end of the winter the field has been cooled to perhaps 4.4-9.9°C (40-50°F). The system is now ready to begin another complete cycle. One now finds that the air-water heat exchanger described earlier and shown in Figure 4.1 is not needed under the new operating mode.

Obviously the cycle will not operate exactly as described due to energy diffusion during the spring and fall. Energy diffusion only changes the temperatures given and not the validity of the proposed operating mode.

4.0 SUMMARY

The brief description of the Detached Earth Tempering concept given here is meant to be only on introduction to the basic concept as first discussed. Later chapters will discuss the computer simulation, the experimental fields, and many refinements of the basic concept. Obviously many changes, improvements, and refinements come from the additional work discussed in later chapters.

REFERENCES

1. "Panel Heating and Cooling Systems," **SYSTEMS HANDBOOK**, American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc., 1980.

2. "Martin Marietta Interactive Thermal Analysis System," Martin Marietta Corporation, Denver, Colorado, 1974.

3. Wright, Scott, "T-Node Thermal Network Simulation Program," Thesis for Masters of Architecture, Georgia Institute of Technology, August 1981.

CHAPTER V

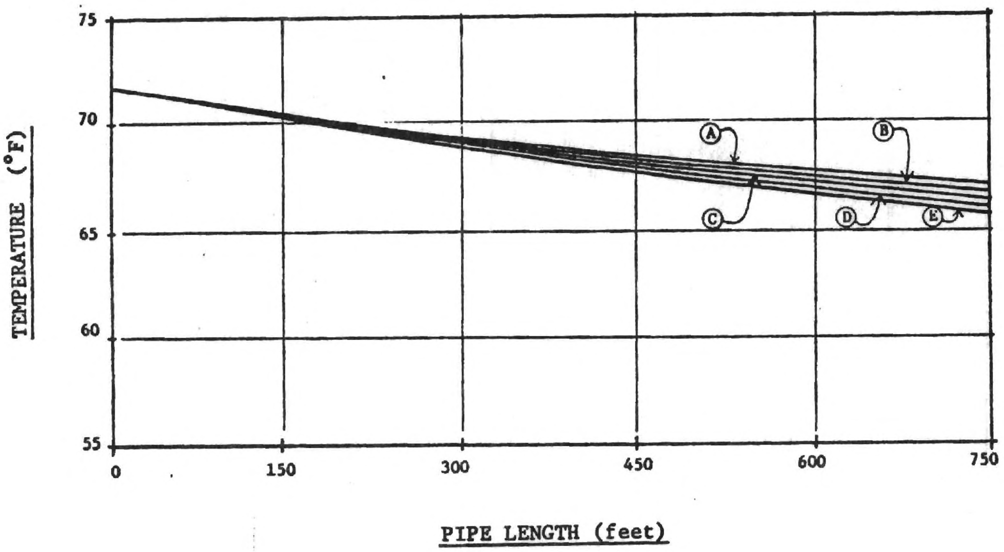
DETACHED EARTH TEMPERING SIMULATION

Although the detached earth tempering concept appeared to have considerable promise, it was felt that a good thermal model of the system must be developed before experimental fields could be put into place. Several of the simulations used in the earth cooling tube analysis were also applicable to the detached earth tempering simulation.

1.0 LINE SOURCE

The line source equations developed for evaluation of the earth cooling tube performance were used to optimize several of the cooling field parameters. One of the firsts questions addressed was whether the thermal conductivity of the tube wall was critical to the energy transfer. Figure 5.1 shows the energy transfer rate is not very sensitive to the tube material, i.e., the thermal resistance of the soil is so great that the low conductivity type materials perform almost as well as the high conductivity materials.

A second question addressed was that of tube diameter. The line source simulations showed that small diameter pipes transferred more energy per sq. ft. of pipe than larger diameter tubes, i.e., a long length of small diameter tube is more effective than a shorter length of larger tube of comparable area, this is illustrated in Figure 5.2. Tube selection then became a matter of ease of installation, cost, minimum bend radius and pumping power. A tube diameter of .031-.038m (1.25 to 1.50") nominal inside diameter was chosen because of low cost, local availability and ability to be bent to radiuses of .46-.61m (1.5 - 2.0 ft.) The initial single level field used conventional



TYPES OF PIPE	HEAT TRANSFER AT END OF RUN
CURVE A = POLYPROPYLENE	TOTAL Q = 9,630 BTU/HR
" " B = PVC	" " " = 9,760 BTU/HR
" " C = ABS	" " " = 10,392 BTU/HR
" " D = POLYETHELENE	" " " = 11,025 BTU/HR
" " E = COPPER	" " " = 11,955 BTU/HR

Figure 5.1 Transfer Fluid Temperature VS Tube Material

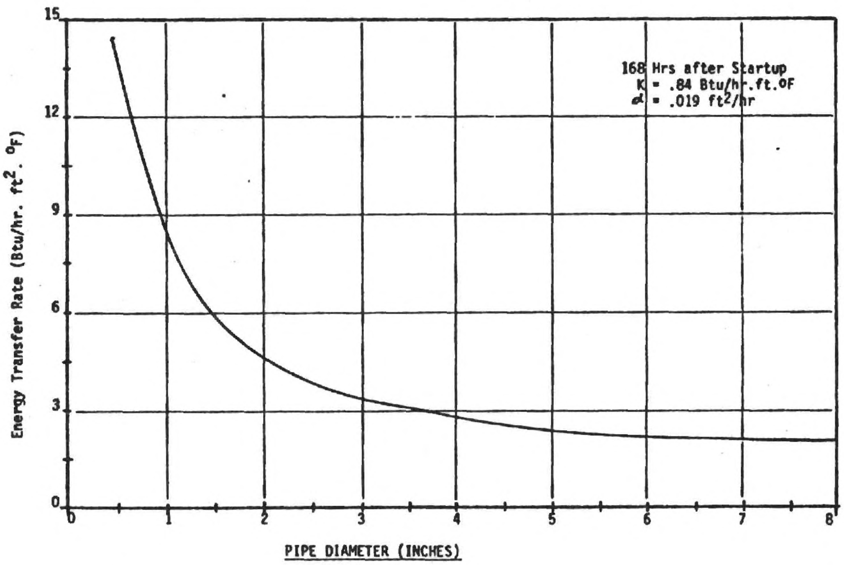


Figure 5.2 Energy Transfer Rate as a Function of Pipe Diameter

polyethylene pipe. The second field used medium density polyethylene 2306 for greater puncture resistance.

It was also necessary to know the soil temperatures at various depths as a function of the time of the year. A number of investigators have published equations for predicting these temperatures [1,2,3]. The equation developed by Kusuda[3] and made popular by Ken Labs[4] was felt to provide accuracy greater than one knew the soil properties. Equation (6.1) below was used to calculate the ground temperatures for Atlanta given in Table V-I and used in the GROCS simulations.

$$T_{x,t} = T_m - (A_s * e^{-x(\pi/365\alpha)^{\frac{1}{2}}}) * \cos\left(\frac{360}{365} * (DAY - DAYO - \frac{x}{2} * (365/\pi\alpha)^{\frac{1}{2}})\right) \quad (6.1)$$

Where: $T_{x,t}$ = Ground temperature at depth x and day d (F)
T_m = Annual mean ground temperature (F)
A_s = Annual surface temperature amplitude (ft)
α = Ground diffusivity (ft sq/day)
DAY = Day of the year
DAYO = Number of days after Jan 1 when surface reaches minimum temperature

It quickly became obvious from the temperature shown in Table V-I that the ground temperatures at reasonable depths (depths which could be reached with a trencher or small backhoe) are too high in late August to provide any useful cooling, even without the increased temperature that would occur when energy was added to the ground from the building. This implied that the ground must be cooled during the winter months to lower temperature than it would normally reach due to conduction from the surface.

2.0 GROCS

A FORTRAN computer model called GROCS (GROund Coupled System)[5] was secured from the Brookhaven National Laboratory. This program had been adapted to work as a subroutine in TRNSYS (a TRaNsient SYstem Simulation program)[6]. GROCS was modified to work with the 10.1 version of TRNSYS which was already being used by Georgia Tech Personnel in several other programs. GROCS was further modified by Georgia Tech Personnel to increase the number of nodes possible so that a finer grain analysis could be made.

Instead of using a very fine grain mesh as is typically used in finite difference solutions to three dimensional energy flow, GROCS solves the heat flow finite difference equations over a system of "blocks" of earth. Each block is a volume of earth whose size and shape are determined from a hand drawn model. Figure 5.3 shows a model for a cooling coil field located 4 ft. beneath the surface.

The block-type model has the following advantages:

1. Useful problems can be solved with relatively short, economical, and simple programs.
2. Adequate accuracy can be obtained since naturally occurring ground inhomogeneities limit accuracy of any model that relies on bulk thermal properties to about 10%.
3. New earth-coupled models can be easily developed and evaluated.

GROCS uses three types of earth blocks. These are called **adjacent blocks, free blocks** and **rigged blocks.** The **rigged blocks** surround the **free blocks** which surround the **adjacent blocks.** The adjacent blocks are free blocks which are adjacent to the heat exchange surface. Fixed blocks have a fixed temperature which is determined by a subroutine called TINTERP. TINTERP reads measured ground temperature data from an input table and interpolates to determine the temperature of the rigged blocks for each month.

Initial free block temperatures are usually input as data. If these temperatures are input as 0 the subroutine TINTERP assigns initial temperatures as a function of depth and month as with the rigged blocks. While rigged block temperatures at subsequent time steps are determined by TINTERP, free block temperatures are determined by their thermal interaction with each other and with the rigged blocks, and by heat inputs to them from the field.

The major approximation which is made by the buried pipe model which is used to simulate the field is the substitution of a flowing plane sheet of fluid for the buried pipe field. This is showns in Figure 5.4. The method for correcting for the approximation was to calculate a correction factor for the area between the field and the adjacent blocks based on the pipe radius and spacing. Reference 5 goes into considerable detail in developing the equation for the reduction or correction factor. The correction factor may be determined by equation 6.2 below.

$$\eta = \left(\frac{\pi h}{s} \right) * \left(\frac{1}{\ell n \left(\frac{s}{\pi R} * \sinh \left(\frac{\pi h}{s} \right) \right)} \right) \qquad (6.2)$$

Where η = correction factor
h = the half-width of the adjacent blocks
s = tube spacing
R = the tube radius
sinh = hyperbolic sin

The accuracy of this approximation was checked by running a model on both GROCS and MITAS (Martin Marietta Interactive Thermal Analysis System)[7] and comparing the results. MITAS is a large thermal network program capable of using a large number of nodes and requiring a time consuming model development and

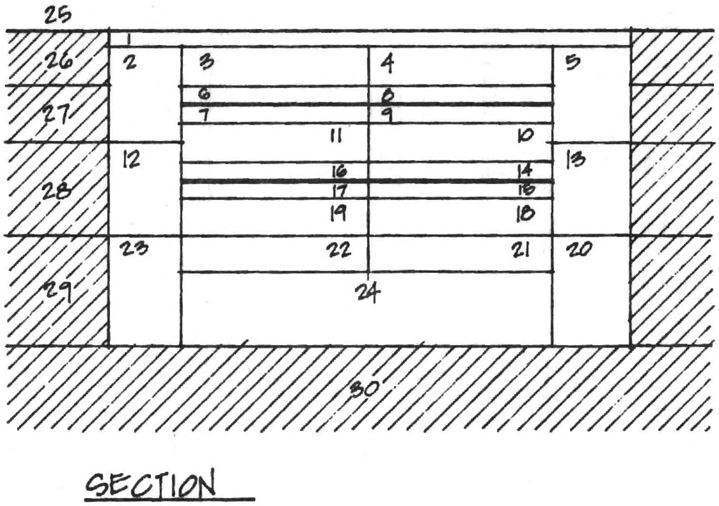


Figure 5.3 Cooling Field Model

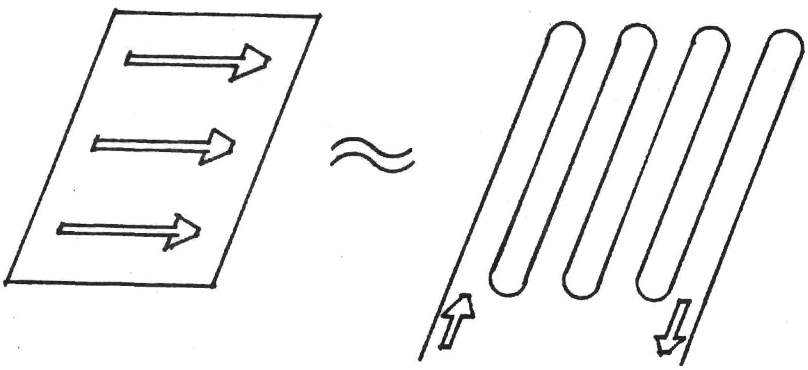


Figure 5.4 Flowing Sheet Approximation of Pipe Array

considerable computer time to run the model. It was determined that the results predicted by GROCS agreed with those predicted by MITAS to well within the 10% limits caused by uncertainties about soil properties and ground inhomogeneities. While GROCS was always run with fewer than 50 blocks, MITAS was run with over 1000 nodes. Appendix I gives a sample GROCS input as well as a sample GROCS/TRNSYS output.

Three basic models were simulated in these tests, a single plane model shown in Figure 5.5, a double level model shown in Figure 5.6, and a double level model beneath a house which is shown in Figure 5.7. Each of the basic models was run through many parametric simulations to determine the effect of different controllable variables. Details of the various models and the results obtained from the parametric studies will be discussed in a later section.

2.1 Modifications to GROCS

A number of modifications were eventually made to the Brookhaven GROCS subroutine as the model sophistication increased. These changes were not made to increase the accuracy of the model but to permit one to either look at things impossible with the standard GROCS or to permit looking at the energy transfer in a little greater detail.

2.1.1 Node Number

The number of nodes permitted in the original GROCS was 20 free blocks 10 adjacent blocks and 8 rigged blocks. This was changed to 40 free blocks, 20 adjacent blocks, and 8 rigged blocks. Theses changes were made to permit us to look at the energy flow in the double level field in more detail than was possible with the standard GROCS.

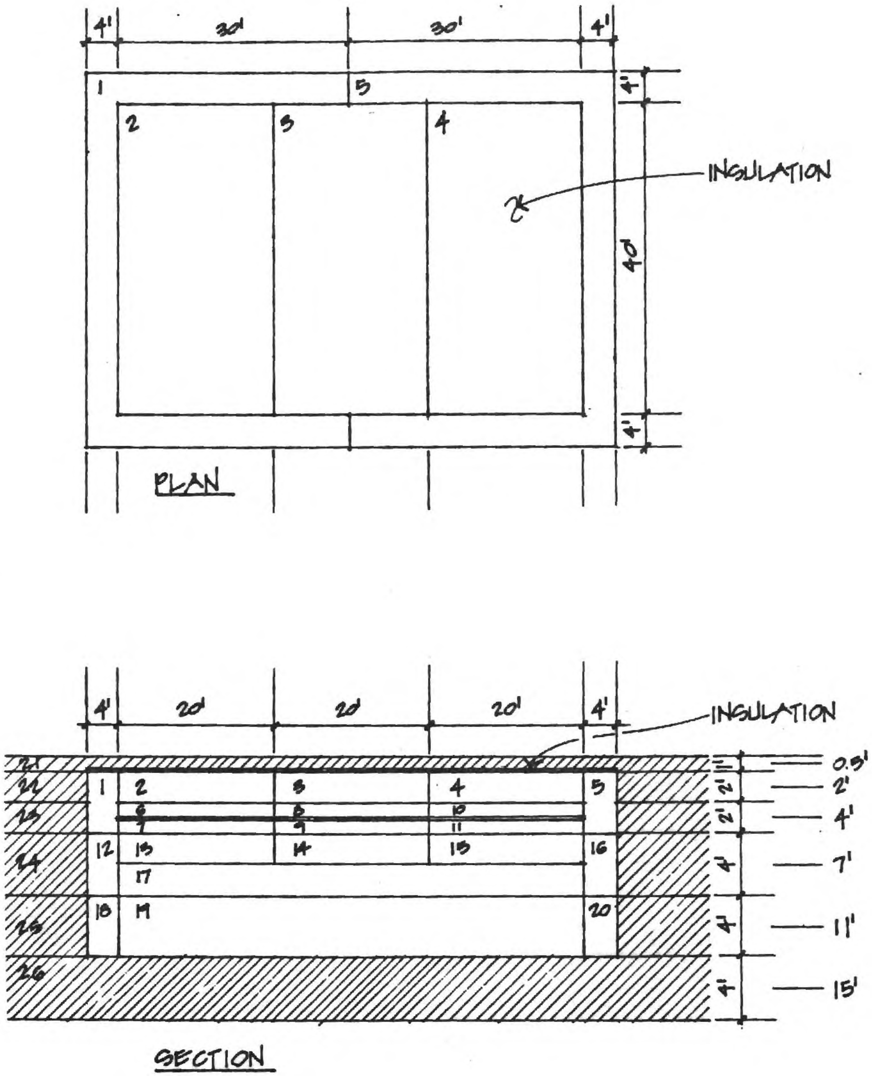


Figure 5.5 Single Level Field Model

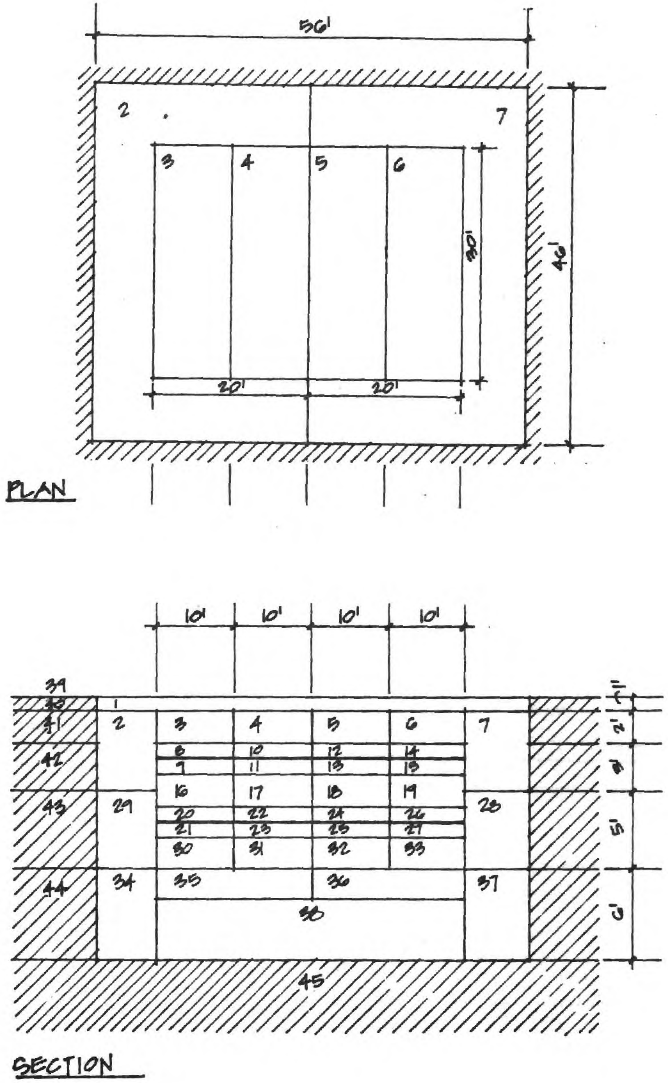
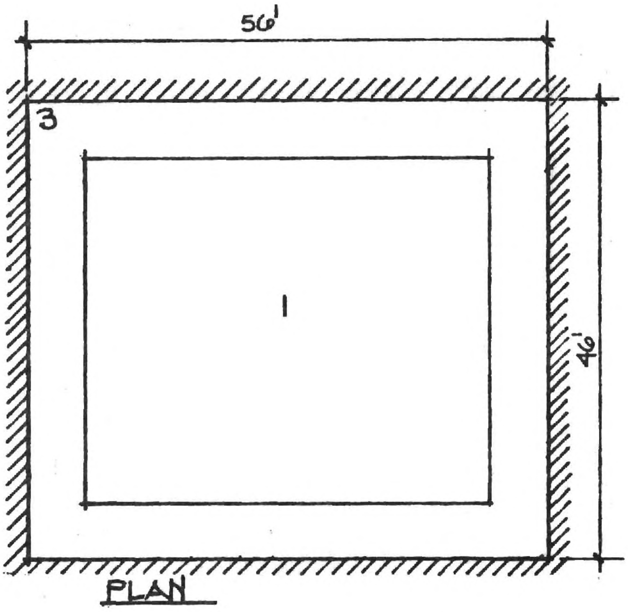


Figure 5.6 Double Level Field Model



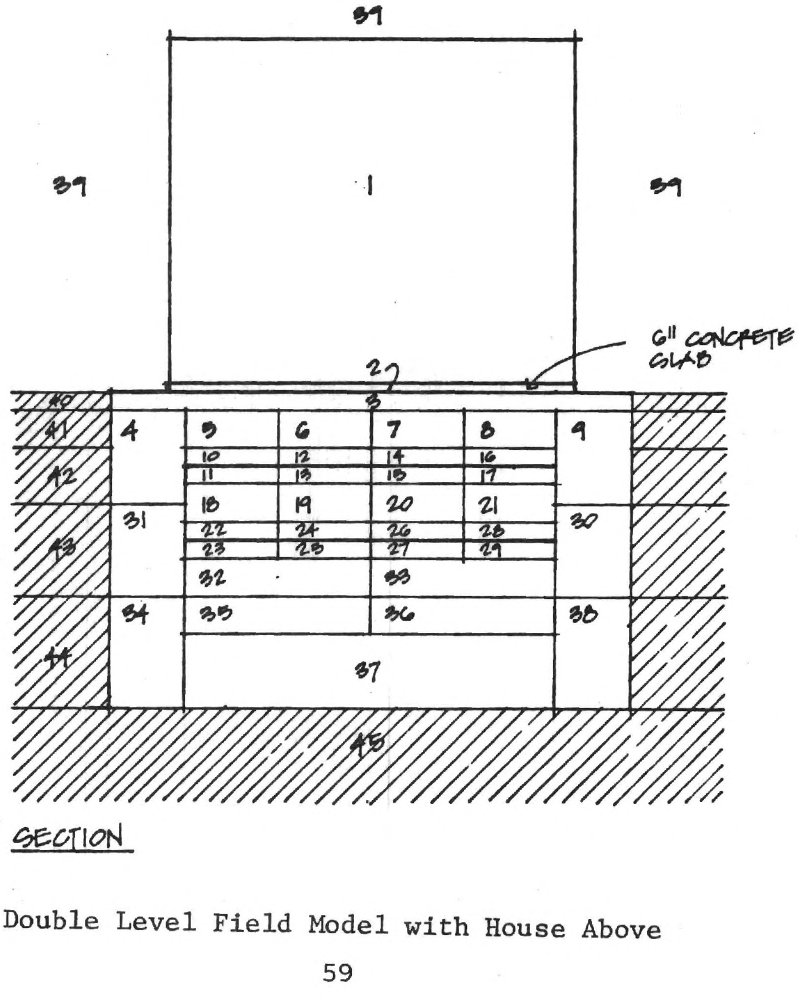


Figure 5.7 Double Level Field Model with House Above

2.1.2 Variable Surface Temperature

When simulations of the double level field beneath the house were begun, it became desirable to have a variable outside temperature so that temperature swings within the house throughout the day and year could be observed. This was accomplished by adding a subroutine called VARST to TRNSYS. VARST calculates a sinusoidal daily and yearly surface temperature which can then be used to vary the temperature of the first rigged block.

The equations used to predict the ground temperature at any depth and any time of the year were modified by superposition of a daily temperature swing as predicted by equation 5.3.

$$Thr = Tave + (\frac{Tswg}{2} * \sin(360 * \frac{hr + 15}{24})) \tag{5.3}$$

Where:

Thr	=	Temperature at time hr	(F)
Tave	=	Average monthly temperature	(F)
Tswg	=	Daily temperature swing	(F)
hr	=	Hours after midnight	(F)

When this equation is superimposed on the ground temperature equation for a depth of zero (surface) one gets equation 5.4.

$$Tsur = Tm - (As * \cos(\frac{360}{365} * (Day - Day0))) + \frac{Tswg}{2} * \sin(360 * (\frac{hr + 15}{24})) \tag{5.4}$$

Where:

Tsur	=	Temperature at time hr	(F)
Tm	=	Average monthly temperature	(F)
As	=	Daily temperature swing	(F)
Day	=	Day of the year	
Day0	=	Days after Jan 1st when minimum temperature occurs	
Tswg	=	Daily temperature swing	
hr	=	Hours after midnight	

The variable surface temperature only became important and was only used when the model being simulated had a field beneath a house and one wished to observe the temperature swings within the house when no auxilliary cooling was used.

3.0 FIELD SIMULATION

Three basic different fields were simulated in these analyses, a single level field at a depth of 1.2m (4 ft.), a double level field with one level at 1.2m (4 ft.) and one at 2.4m (8 ft.) and finally a double level field similar to that above but located under a house and coupled to the house through a slab floor. Parametric studies were conducted on each basic type to determine the optimum tube spacing, the optimum insulation thickness and the optimum energy extraction rate.

3.1. Single Level Field

The single level field was located at a depth of 1.2m (4 ft.) and was limited to a plan area of 223m^2 (2400 sq. ft.) (12.2 - 18.3m) (40' x 60'). Parametric studies were conducted to determine optimum tube spacing, optimum insulation thickness and optimum insulation placement. These simulations showed that horizontal placement of the insulation beyond the edge of the coil field was better than a vertical placement down along the edge of the field. The 1.2m (4 ft.) extension beyond the edge of the coil field was selected as the optimum distance because insulation cost was increasing much faster than the energy loss was decreasing.

Figure 5.5 shows the basic model used in the single level field simulations. Notice that the same basic model is used for all single level simulations. One only needs to vary the area between adjacent blocks to take into consideration any insulation which may lie between the blocks. The model was originally set up for an energy extraction rate of 1759W (6000 Btu/hr) which remained constant until the field was no longer capable of supplying energy at that rate. The fluid flow rate through the field was adjusted to keep the energy extraction rate constant until the mass flow rate reached 1814 Kg/hr (4000 lbs/hr). This required the flow rate to be continually adjusted throughout the simulation. Flow rate would be low during the initial hours when the field temperature was low. Flow rate would then increase until it reached a maximum of 1814 Kg/hr (4000 lbs/hr), at which point the field could no longer supply 1759W (6000 Btu/hr) and the cooling capacity would begin to drop.

Figure 5.8 shows the effect of insulation on the time the field is capable of supplying the 1759W (6000 Btu/hr). Notice the significant improvement when going from no insulation to R10 insulation and the still further improvement with further increases in insulation.

These simulations indicated that while the single level field would be capable of supplying a significant percentage of the total cooling load of a well insulated house passively, the quantity and thickness of insulation required was significant because the plan area was large and the greatest losses were through the insulation to the surface. This suggested that a double level field of less plan area might offer a significant improvement.

3.2 Double Level Field

The single level field simulations suggested that a double level field would not only perform better but would also cost less to construct. Figure 5.6 shows the basic model used for the double level field. Since the field depth was twice that of the single level field, the plan area of the double level field was reduced to 111.5m^2 (1200 sq. ft.) (30' x 40'). The insulation was extended 2.8m (8 ft.) beyond the edge of the pipe field, i.e., equal to the greatest depth.

The GROCS subroutine was modified to increase the number of earth blocks possible, for simulation of the double level field so that the energy flow paths could be more closely followed. Parametric runs similar to those conducted on the single level field were also conducted on the double level field. Since the double level field used .031m (11/4 in.) nominal inside diameter pipe, the pipes were placed on .9m (3 ft.) centers rather than 1.22m (4 ft.) as had been used on the single level field. This permitted 304.8m (1000 ft.) of pipe to be positioned in the two level.

Maximum energy extraction rates and maximum fluid mass flow rates were kept identical to those used in the single level field simulations. Figure 5.9 shows that the double level field does indeed perform significantly better than the single level field, with the uninsulated performance of the double level field being better than the R10 insulated performance of the single level field. R10 insulated performance of the double level field is again much better than the uninsulated performance. While some improvement is evident with perfect insulation between the field and the surface, the improvement is not nearly as great as was experienced with the single level field. Notice that there is little difference in performance of the single and double level fields when both have perfect insulation between the field and the surface.

3.3 Double Level Field Beneath A House

The plan area of the double level field has been decreased to the point where one could now begin to consider putting the field beneath the house rather than in an open field. This serves several important purposes. First, it decreases the impact of high solar imput to the field surface. Second, energy gain to the field through the top surface serves to cool the residence rather than being lost to the ambient air.

Figure 5.7 shows the basic model used for the field beneath the house. Notice that it is identical to the double level field discussed above, with the addition of a concrete floor and house above the field. It was felt that a passive approach such as this would be most effective when used with a very well insulated house. A 148.6m^2

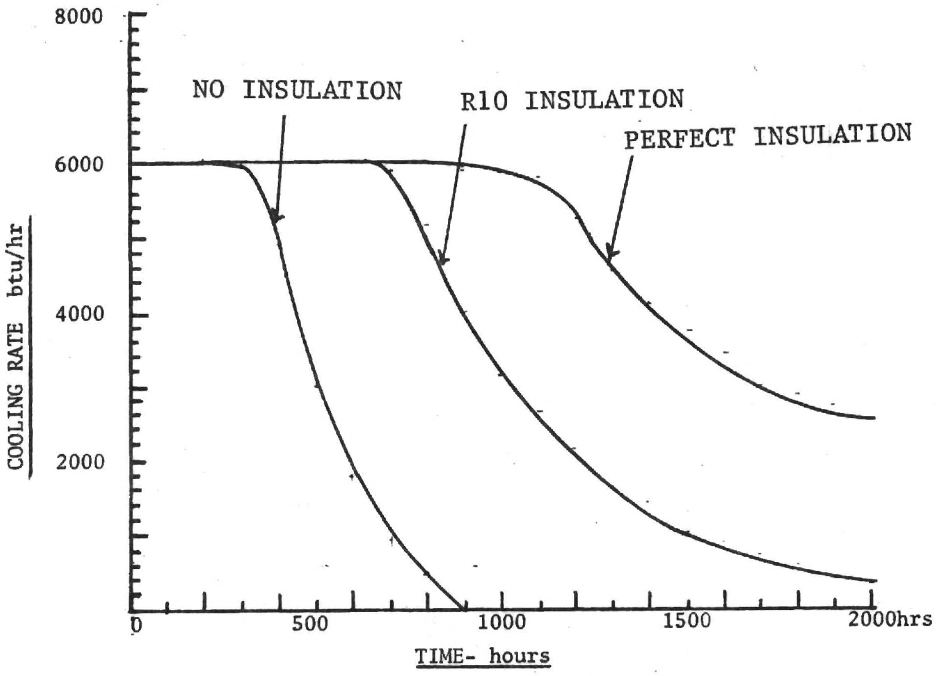


Figure 5.8 Cooling Capacity of Single Level Field

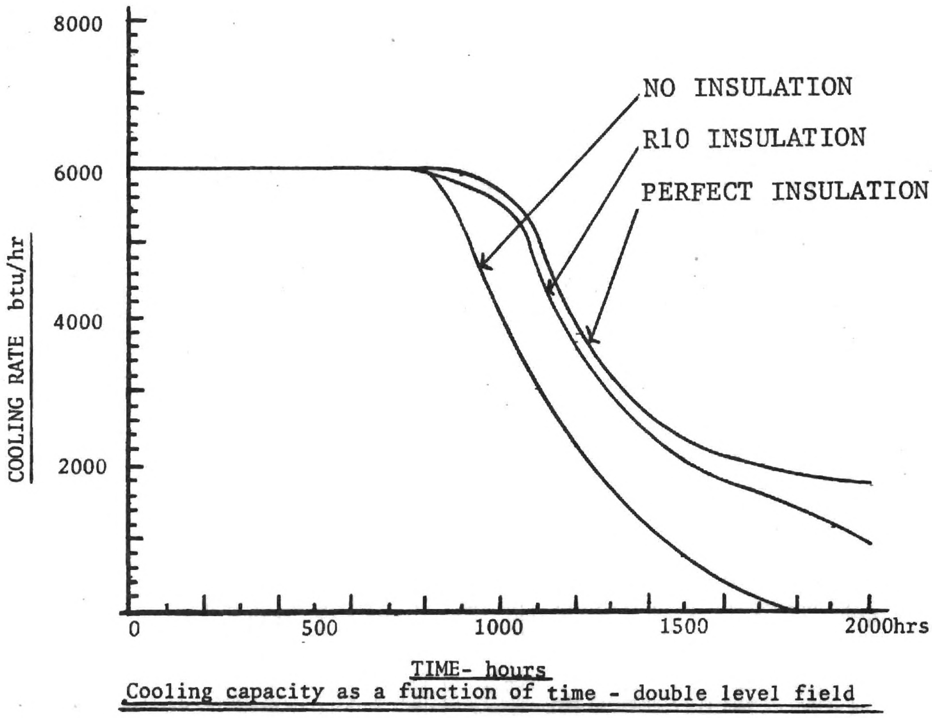


Figure 5.9 Cooling Performance of Double Level Field

(1600 sq. ft.) house with a UA of 184.6W/°C (350 Btu/hr F) was chosen for the simulations.

Simulations were run for both 1759W (6000 Btu/hr) and 879W (3000 Btu/hr) energy extraction rates. Figures 5.10 and 5.11 show the results from both of these cases. One could say that the lower extraction rate would obviously continue to work longer than the higher rate and that it would also have worked longer for both the single and double level fields discussed above, and one would have been correct. What prompted the simulations using the lower extraction rate was a study of the floor and ambient air temperatures within the house without using the cooling capacity being extracted through the tubing. It was found that the house could be maintained at a comfortable temperature without using any of the energy being extracted through the tubes, i.e., the field coupled to the house through the concrete floor provided sufficient cooling to meet the cooling load without any auxilliary cooling. It appears from these initial simulations that the system must be carefully designed to prevent overcooling during the early part of the summer.

Simulation of the fields beneath the house have only been carried sufficiently far to indicate the potential. Obviously considerably more simulation is required to optimize this approach. It appears from the initial simulations that the buried pipe field may only be needed during the charging cycle and possibly as a loop to move energy from the upper level to the lower level. All cooling to the house would be accomplished through the floor and would be regulated by the rate at which energy is moved from one level to the other in the cooling field.

The effectiveness of this approach can be indicated best by looking at the temperature swing within the house in early July after the cooling system has been providing cooling for 1900 hours. This simulation is for a house with a UA of 184.6 (350) located over a double level field with .051m (2 in.) Dow SM insulation located .3m (1

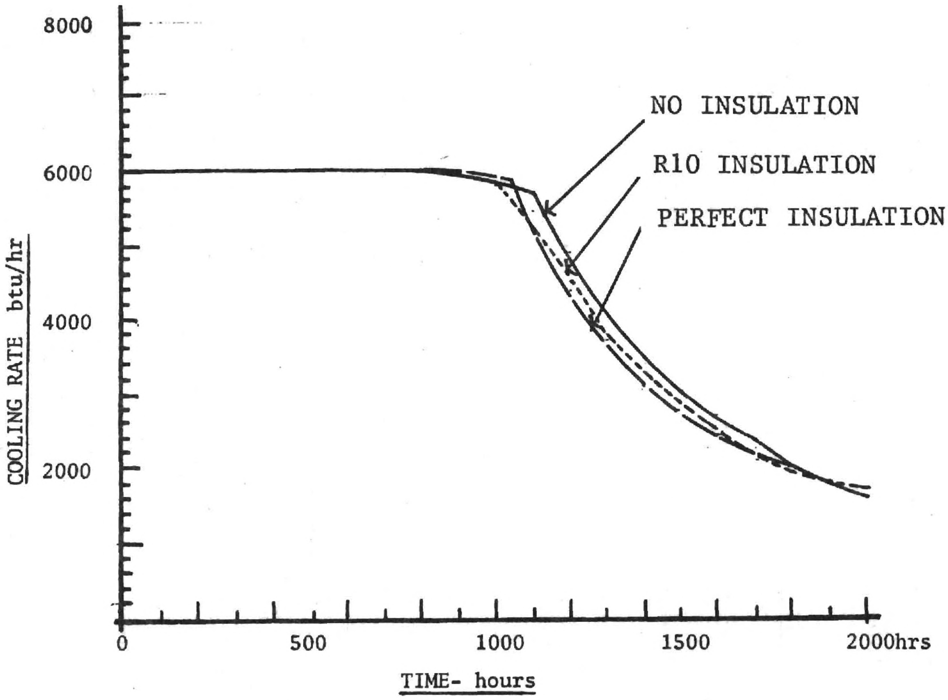


Figure 5.10 Cooling Performance of Double level Field with House

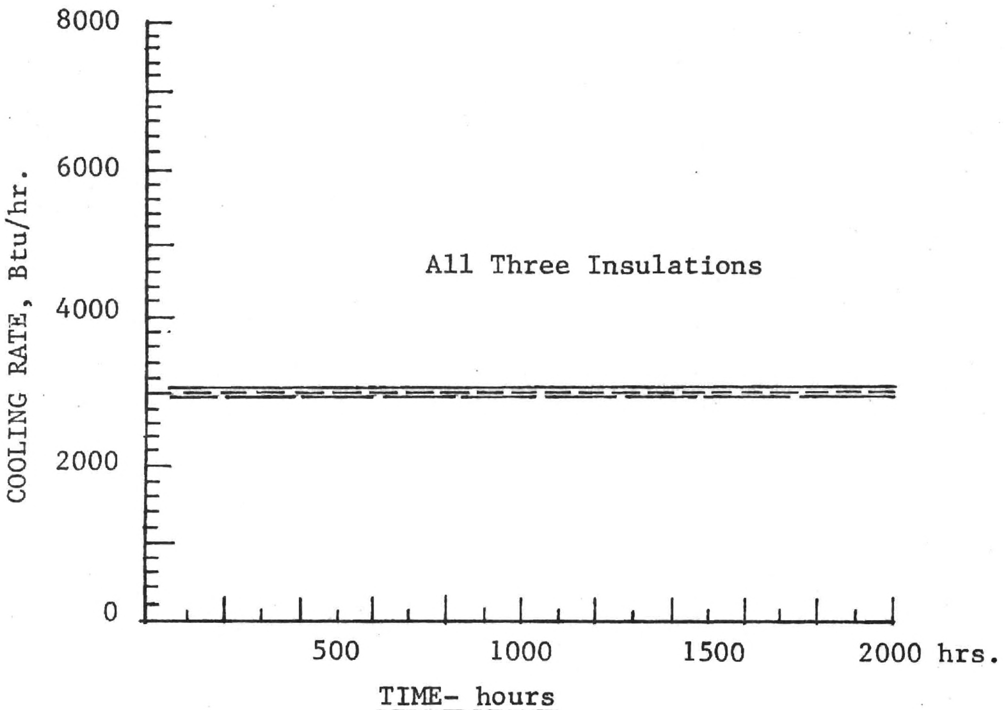


Figure 5.11 Cooling Performance of Double level Field with House

ft.) below a .15m (6 in.) concrete floor. Figure 5.12 shows that the exterior ambient air temperature varies from 23.3C to 32.8C (73.9 F to 91.1 F) while the air temperature within the house varies from 23.8C to 24.8C (74.9 F to 76.7 F) and the floor temperature varies from 23.6C to 23.9C (74.4 F to 75.1 F).

A word of caution should be inserted here. The particular example given would not be a viable design for a residence because the house interior temperature would vary from 15.7C to 16.0C (60.33 F to 60.85 F) in early May. The example was given to indicate the potential of this approach. Since insufficient simulations have been conducted on this system, it is not optimized and suffers from too much cooling during the early summer. It is believed these problems can be overcome by varying the depth of the fields and varying the insulation location. Obviously considerable computer simulation will be required to optimize the design. It will also have to be optimized for each different climate and for each different house to reach its full potential.

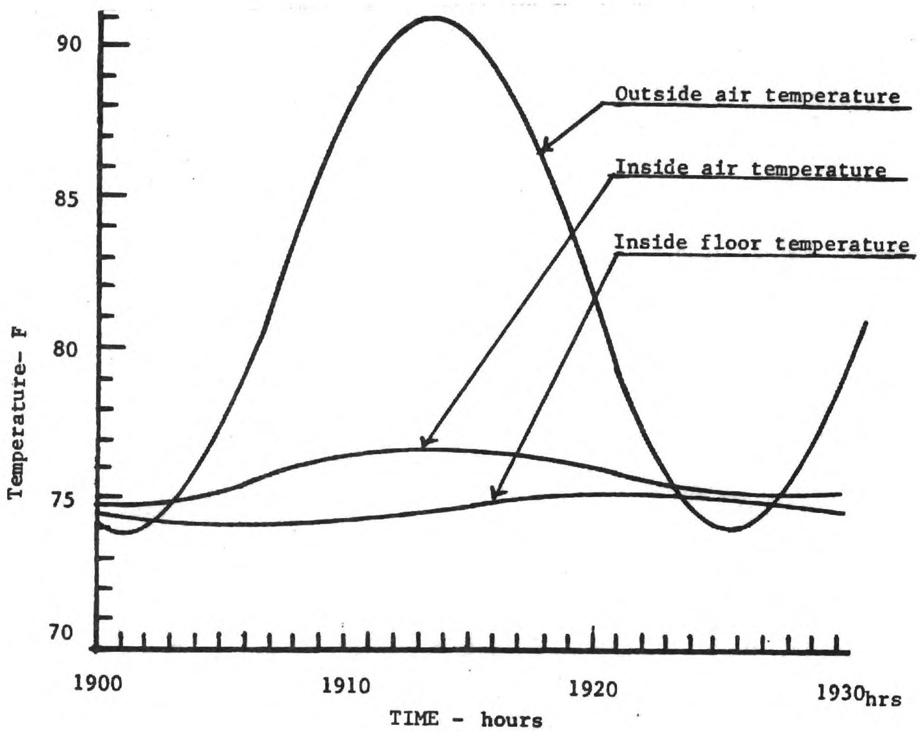


Figure 5.12 Thermal Performance of House Over Field

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CHAPTER VI

DETACHED EARTH TEMPERING EXPERIMENTAL

1.0 GROUND TEMPERATURES

Although the ground temperature prediction equations are considered to be quite accurate, it was decided to sink two forty 12.2m (40) ft. wells to permit ground temperatures to be monitored. Figure 6.1 shows the well locations relative to the field location. The well location was chosen so that one well would be located vertically through the field and one would be located about 15.24m (50) ft. west of the field hole and sufficiently far from the field to not be affected by the energy transferred to the ground by the field.

Figure 6.2 is a cross section through one of the wells showing thermocouple location. The thermocouples are positioned in a sand filled PVC tube with each thermocouple bead projecting through a small hole in the tube and being bonded to the exterior of the tube. Since the PVC has a conductivity of only 1/10 that of soil, the tube should not affect the thermocouple measurements. Once the thermocouple tube was located vertically in the hole, the hole around the tube was backfilled with a 50/50 mixture of bentonite and cement. This mixture was chosen because it has a thermal conductivity not significantly different from that of the soil, it flows readily, and should not subsequently develop cracks which might affect soil temperatures. Figure 6.3 shows a detail of the PVC tube at one of the thermocouple locations.

Surprisingly, both wells hit water at 6.1-7.0m (20-23) feet. This was surprising because all the soils experts in this area said that one should not hit water in the Atlanta area at depths less than 76.2m (250) feet. The particular site chosen for the field is an old stream bed with a mixture of silt and clay as the predominant soil type. Apparently, considerable water remains in the soil despite the storm sewer shown in Figure 6.1.

While it would have been desirable to have a continuous measurement of soil moisture in the vertical wells as well as throughout the field, lack of suitable soil moisture probes precluded the installation of moisture measuring equipment. It had been anticipated that moisture measuring probes developed by the U.S. Forest Products Laboratory at Athens, Georgia, would be satisfactory for the soil moisture measurements. Mr. J. E. Duff of the Forest Products Laboratory advised that these probes are not satisfactory for soil moisture measurements. After talking with Mr. Duff, Dr. Tom Bligh of MIT, Dr. Phil Metz of Brookhaven National Laboratory, Dr. Jim Hartley of Georgia Tech and with Georgia Power personnel, it was concluded that no satisfactory remote soil moisture measurement device existed within our operation and budget parameters.

Soil samples were taken every five feet during the drilling operation. These samples were weighed, dried and reweighed. This permitted us to make an initial soil moisture determination and with the basic soil type and moisture content estimates of soil conductivity, diffusivity, and heat capacity were made. Table VI-I gives the soil weight and thermal properties as determined by these tests.

Soil temperature measurements taken during the first fall were very close to those predicted by the equations, although temperatures measured in hole 2 were several degrees lower than predicted. Hole 2 is located adjacent to a large elm tree and is located on the north side of this tree. It appears that the tree is effective in lowering adjacent soil temperatures. It appears this effect is a combination of reduced solar input and evaporative cooling adjacent to the tree. Temperature measurements at Hole #1 during the summer months began to show a significant deviation from those

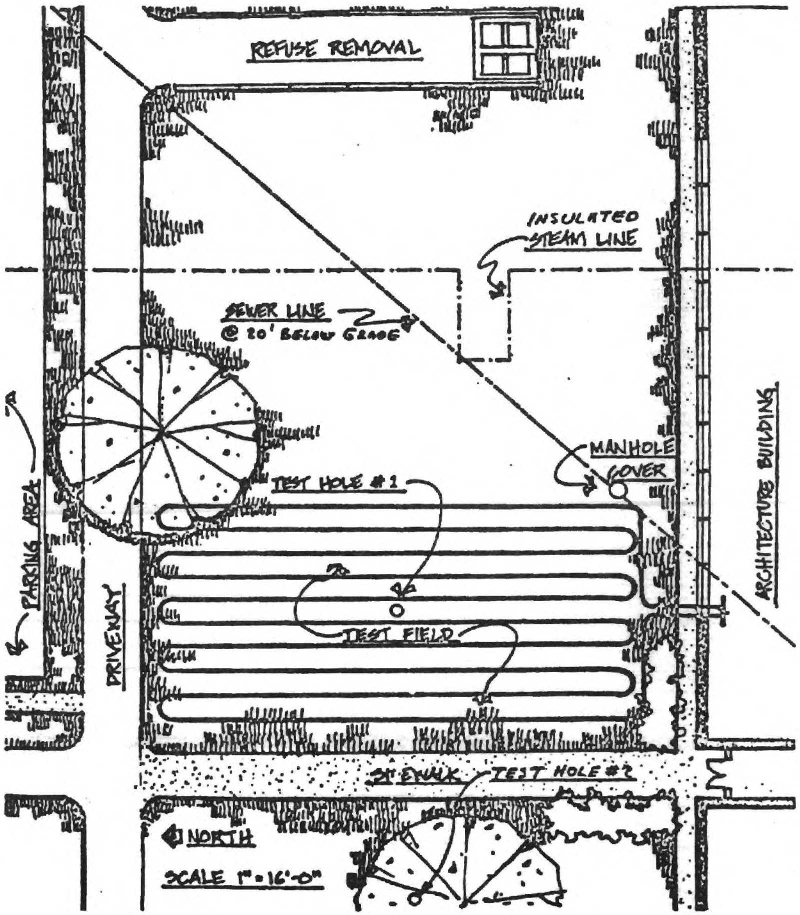


Figure 6.1 Plan of Field Showing Wells

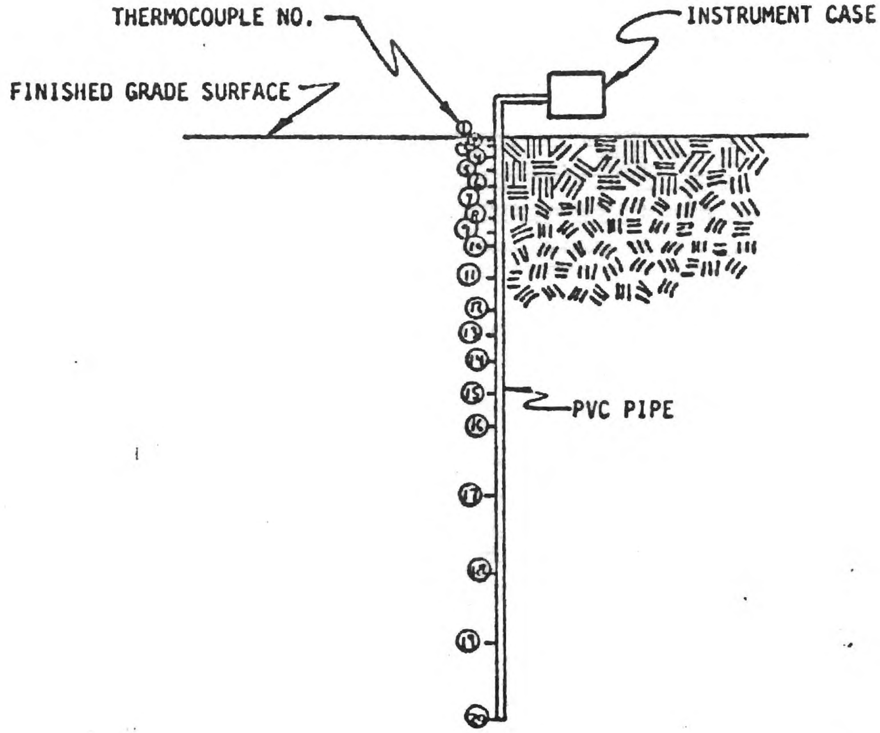


Figure 6.2 Cross Section of Temperature Well

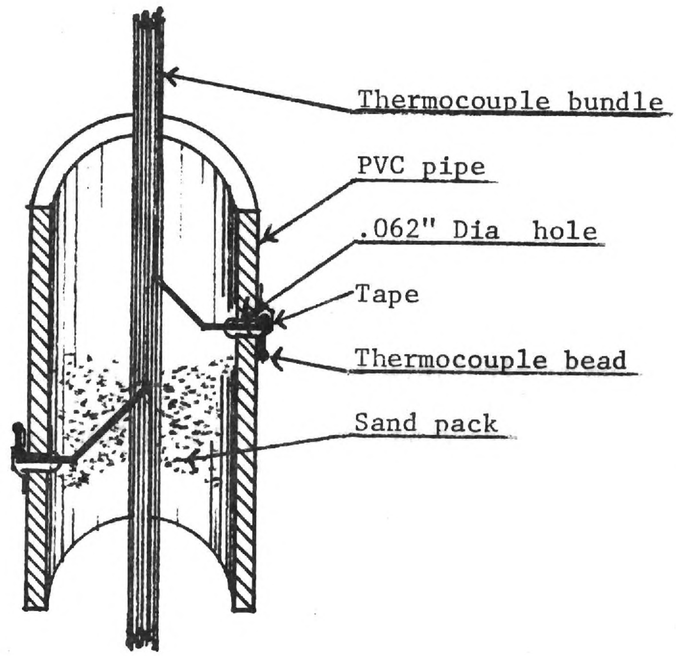


Figure 6.3 Section of Ground Temperature Probe

TABLE VI-I Soil Properties from Test Holes 1 and 2

SAMPLE NUMBER	DEPTH Ft	DRY DENSITY Lbs/Ft^3	MOISTURE CONTENT % Dry Wt.	THERMAL CONDUCTIVITY BTU Ft/Ft^{2}hr. °F	HEAT CAPACITANCE BTU/Ft.^{3} °F	THERMAL DIFFUSIVITY Ft^2/hr.
B.1.1	3.5 -5.0	83.1	12.1	.556	24.5	.023
B.1.2	8.5-10.0	61.4	27.1	.609	27.3	.022
B.1.3	13.5-15.0	80.8	22.1	.864	31.9	.027
B.1.4	18.5-20.0	86.1	25.2	1.052	36.6	.029
B.1.5	23.5-25.0	81.7	35.0	1.052	42.8	.025
B.1.6	28.5-30.0	79.2	35.0	1.015	41.5	.025
B.1.7	33.5-35.0	77.3	35.0	.970	40.5	.024
B.1.8	38.5-40.0	81.8	35.0	1.054	42.8	.025
B.2.1	3.5 -5.0	67.4	17.9	.590	23.8	.025
B.2.2	8.5-10.0	69.1	13.7	.476	21.5	.022
B.2.3	13.5-15.0	71.1	23.6	.727	29.1	.025
B.2.4	18.5-20.0	97.3	16.6	1.043	33.0	.032
B.2.5	23.5-25.0	84.2	35.0	1.167	44.1	.026
B.2.6	28.5-30.0	78.0	35.0	.969	40.8	.024
B.2.7	33.5-35.0	76.5	35.0	.937	40.0	.023
B.2.8	38.5-40.0	89.6	35.0	1.281	46.9	.027
Old Snow				.16	14	.01
Dry Sand				.1	19	.005
Wet Sand				1.0	25	.04

predicted. The equations used to predict the soil temperatures assumes a sod ground cover. Hole #1 was located in an open field with little or no ground cover. Surface temperatures as high as 56.7°C (135° F) were measured when the ambient temperature was less than 37.4°C (100° F). Surface temperatures at Hole 2 were consistently below the ambient temperature during the summer months. Temperatures in both holes at depths below the water level were identical and were about the 17.5°C (63° F) predicted by the equations.

Figure 6.4 shows the temperature distribution from the surface to a depth of forty feet for both the test hole (that located in the cooling field) and the reference hole located about 17m (56 ft.) to the west. Notice the significantly lower temperatures in the reference hole for both dates than for the test hole. This difference results primarily from the reference hole being located in the shade of a tree while the test hole was in an open field with no ground cover. This illustrates the importance of

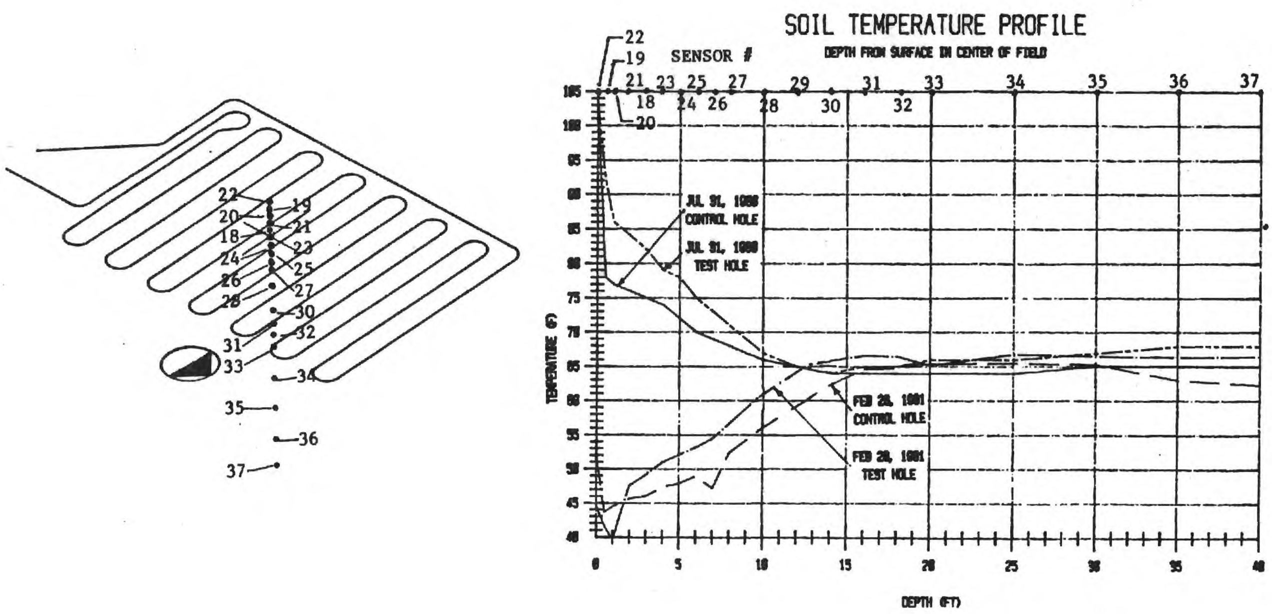


Figure 6.4 Temperature Distribution in Test Holes

ground cover for ground used for passive cooling purposes. The lower temperatures at the surface and one foot depth for the test hole in February results from the insulation at the one foot depth decreasing energy flow from the lower depths.

2.0 EXPERIMENTAL COOLING FIELDS

As the computer simulations on the single level field began to show promise, plans were begun to construct a single level field based on the optimum configuration developed during the simulations. This field was constructed and evaluated during th winter and summer of 1981. The field is described in detail below along with the performance measured.

While the single level field was being consructed and evaluated, the computer simulations continued. These simulations soon began to show the significant increase in performance that one could realize through the utilization of the double level field. The simulations also showed the significant potential of the double level field located beneath the house. Since project funds were almost exhausted,construction of a field beneath a house was out of the question.

When the single level field began to develop leaks due to plastic creep caused by large rocks pressing on the low-density plastic pipe (rocks which should have been removed when the field was backfilled), it was decided to replace the field with a double level field. Georgia Power provided sufficient funds to replace the field. Details of the double level field and the precautions taken to prevent the leakage problems which had plagued the single level field are discussed below.

2.1 Single Level Field

The single level field consisted of 213.4m (700 ft.) of .04m (1½ in.) nominal diameter black polyethylene tubing installed at a depth of 1.2m (4 ft.). The original plans had been to lay the pipe in a .15m (6 in.) wide trench made by a powered trencher. This plan was soon abandoned after it was discovered that the field was located in a site where an old road had previously run. Both the curbstone and the cobble stones had been left in place when the road was covered. These stones, some of which weighed 136.3-181.8Kg (300-400 lbs.), proved to be an insurmountable obstacle to the trencher.

The trencher was abandoned and a large bulldozer was used to excavate the site to the four foot depth. Once the site was excavated to the four foot depth, the field was laid out and the site backfilled with three feet of soil that had been removed from the site. The bulldozer was used for backfilling with some care being exercised to minimize the number of rocks which were put back into the field. Later events showed that insufficient care had been used in removing the rocks. These rocks eventually led to considerable problems with tube leakage.

The site was left for four weeks to permit the soil to settle and pack. After the four week settling period, the site was hand-leveled with rakes and covered with .05m (2 in.) of Dow SM extruded polystyrene rigid insulation. The insulation was extended 1.2m (4 ft.) beyond the edge of the coil field to minimize energy gain around the edge of the insulation. The insulation was covered with a .15mm (6 mil) polyethylene film to prevent water migration through the field due to rains.

The field was finally covered with a one foot layer of soil. This was packed and planted with a mixture of rye and fescue grass. Every effort was made to provide a good sod cover to minimize solar heating of the soil above the field.

As previously mentioned, the field was located so that one of the previously installed 12.2m (40 ft.) soil temperature wells was located midway between the serpentine field coils. The field was heavily instrumented with premium grade copper-constantan thermocouples. Figure 6.5 shows the sensor location within the field.

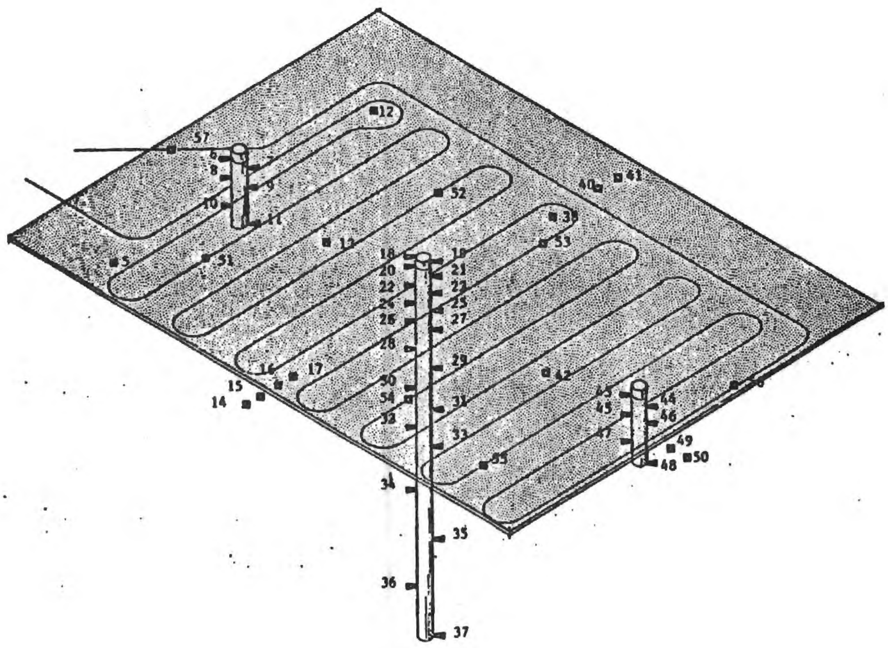


Figure 6.5 Sensor Locations Within Cooling Field

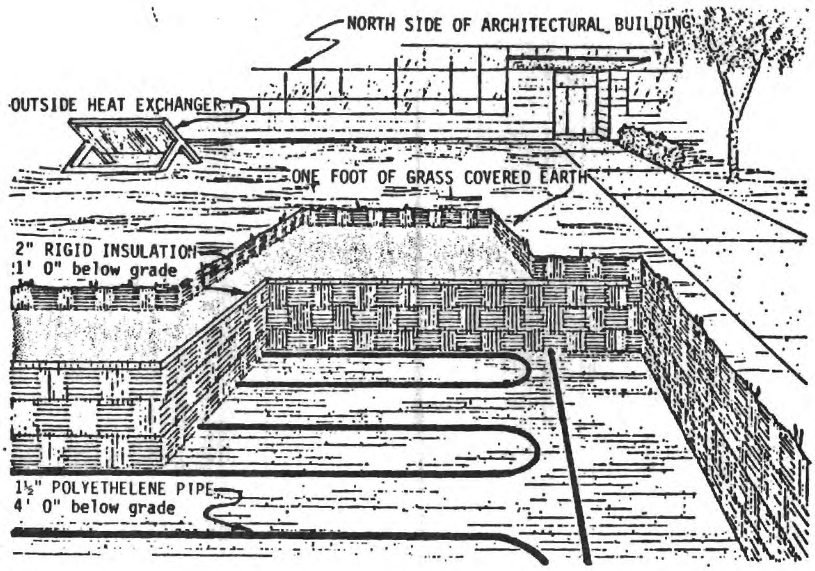


Figure 6.6 Section Through Cooling Field

Figure 6.6 is a section through the field showing the serpentine field, the insulation, and the surface.

2.1.1 Performance Evaluation

Since a radiatively cooled house was not available for evaluation of the field performance, this evaluation was divided into two separate parts so that the effects of each could be evaluated. A load simulator was used to measure and evaluate the performance of the field and determine how well energy could be stored and extracted from the field. A separate test facility was built to evaluate the radiant cooling effect of wall panels.

The building load simulator consisted of a 20478 Btu/hr (6kw) electrical circulation heater whose power output could be controlld through a programmable load controller. Figure 6.7 gives a schematic of the equipment used to simulate the load on a house. Figure 6.8 is a flow schematic showing how the flow changes direction depending on whether the field is being charged or discharged.

Actual measured electrical consumption of a Georgia Power "Good Cents" house located in Columbus, Georgia, was converted to hourly sensible cooling loads for each of the cooling months using the seasonal EER given by the manufacturer of the home's air conditioner. These monthly load profiles are given in Figures 6.9 - 6.13 in both Btu/hour and kilowatts. Only sensible loads are used because the radiative cooling method employed in the Detached Earth Tempering Concept is not capable of carrying the latent loads. Sensible loads were estimated by dividing the total hourly cooling load by 1.3.

The daily load profile for a given month was programmed into a Research Incorporated Model 73211 Micro Data Trak load programmer which controlled a Research Incorporated Model 63911 process controller with a 40 amp solid state switch.

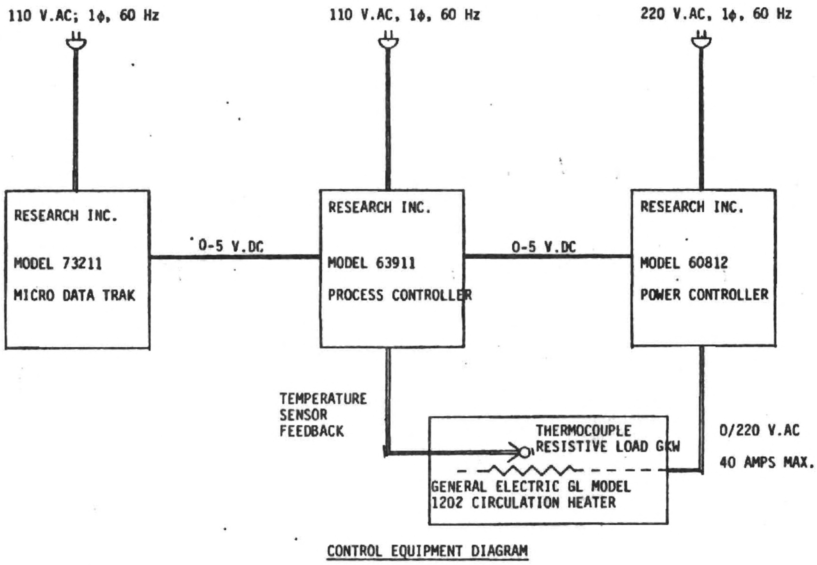
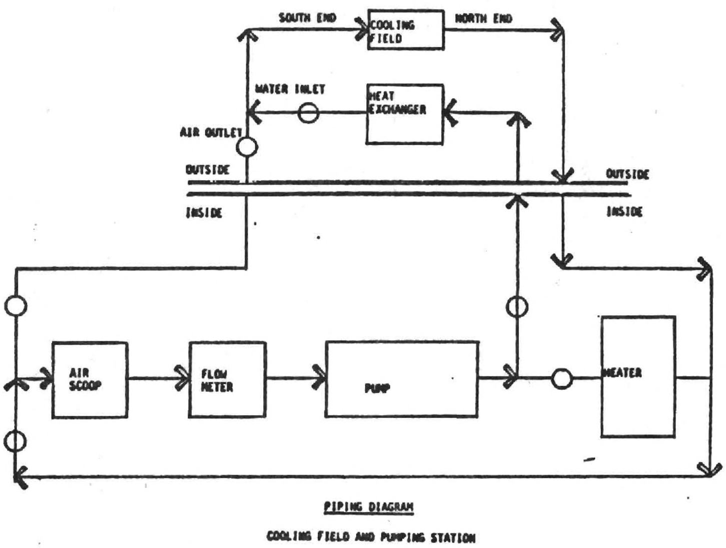


Figure 6.7 Building Load Simulation Schematic



WINTER MODE - CHARGING

Figure 6.8 Cooling Field Flow Diagram

The solid state switch varied the power going to a General Electric 220 volt, 20478 Btu/hr (6 kw) electric circulation heater according to the programmed load.

Water was circulated through the buried field and through the water heater before returning to the field. The field was considered to be capable of passively cooling a building until the water temperature coming from the field rose above 22°C (72° F). The choice of 22°C (72° F) was based on preliminary radiant cooling simulations which showed that comfort would decrease at radiator temperatures much above 22°C (72° F).

Table VI-II shows the monthly sensible cooling load of the house, the cooling load carried by the field and the percentage of the sensible load carried by the field. While the percentage carried passively might at first seem low, one must go back to the start of the field charging to get a true picture of the cooling potential of this concept. Due to program scheduling it was necessary to install the field during the record heat wave being experienced by Atlanta in August of 1980. This resulted in the ground at the field four foot depth being exposed to direct solar radiation and much higher ambient temperatures than the four foot depth would normally experience. To minimize settling, the field was left exposed for over four weeks after it had been backfilled to one foot below grade. Unfortunately, this permitted all of the soil within the field and around the field to reach much higher temperatures than it normally would have reached. Again, due to scheduling, the field was insulated with 5.08cm (2 in.) of Dow extruded polystyrene rigid insulation in late September when the field and adjacent ground were very hot. The insulation trapped this heat within the field and minimized conductive loss to the surface as the ambient temperature began to decrease.

The remainder of the charging circuit, the load simulator, and the control strategy could not be installed until the field was in place, insulated, and the instrumentation completed. The field was not ready to begin charging until the first

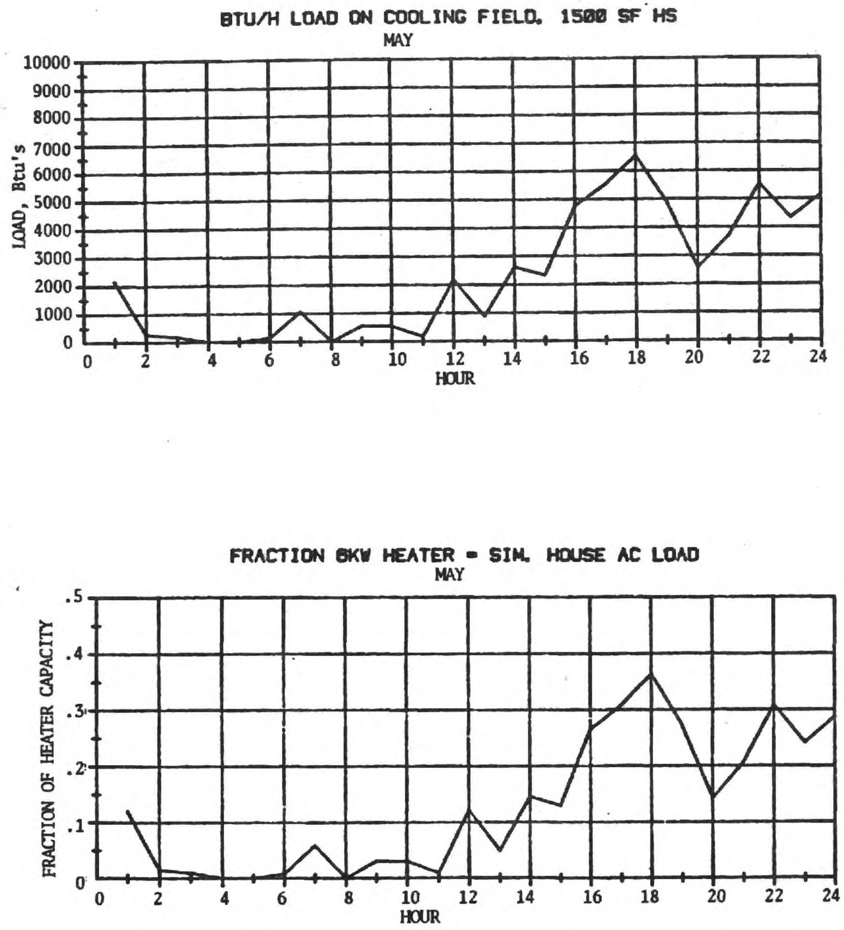


Figure 6.9 Building Load Profile - May

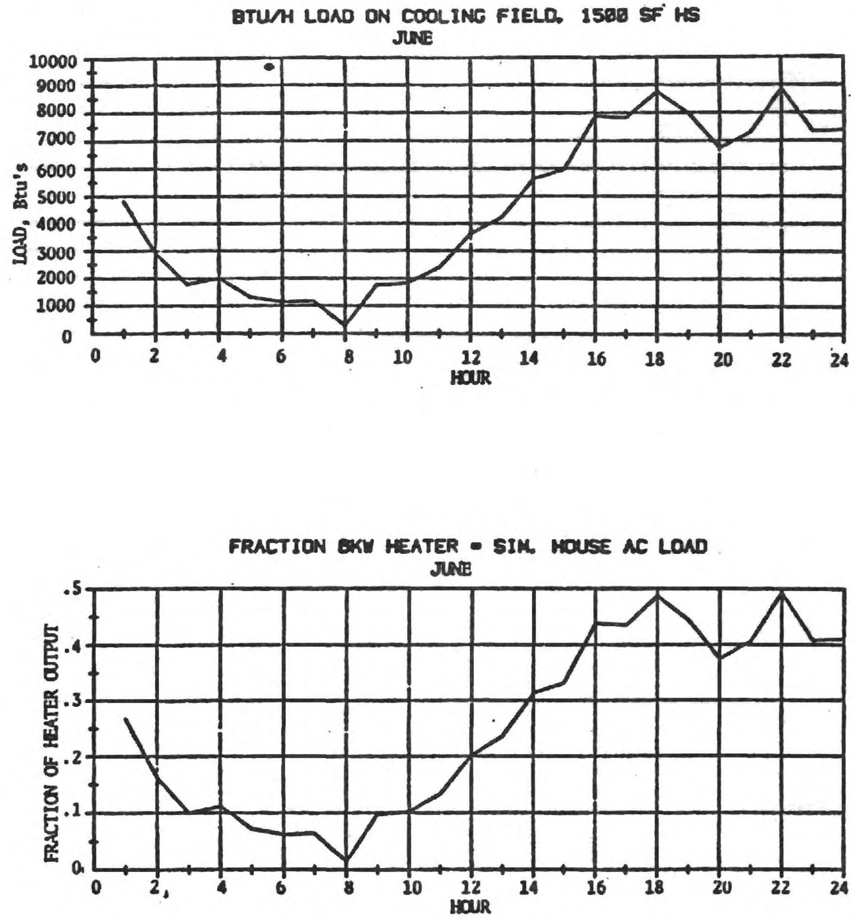


Figure 6.10 Building Load Profile - June

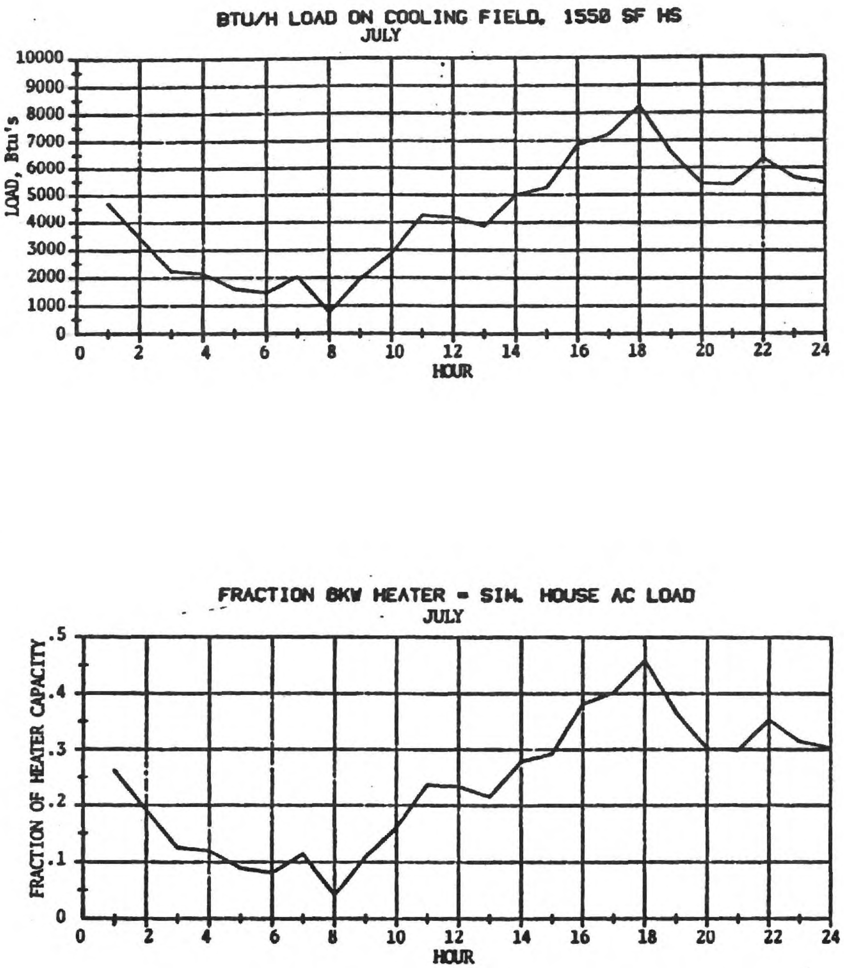


Figure 6.11 Building Load Profile - July

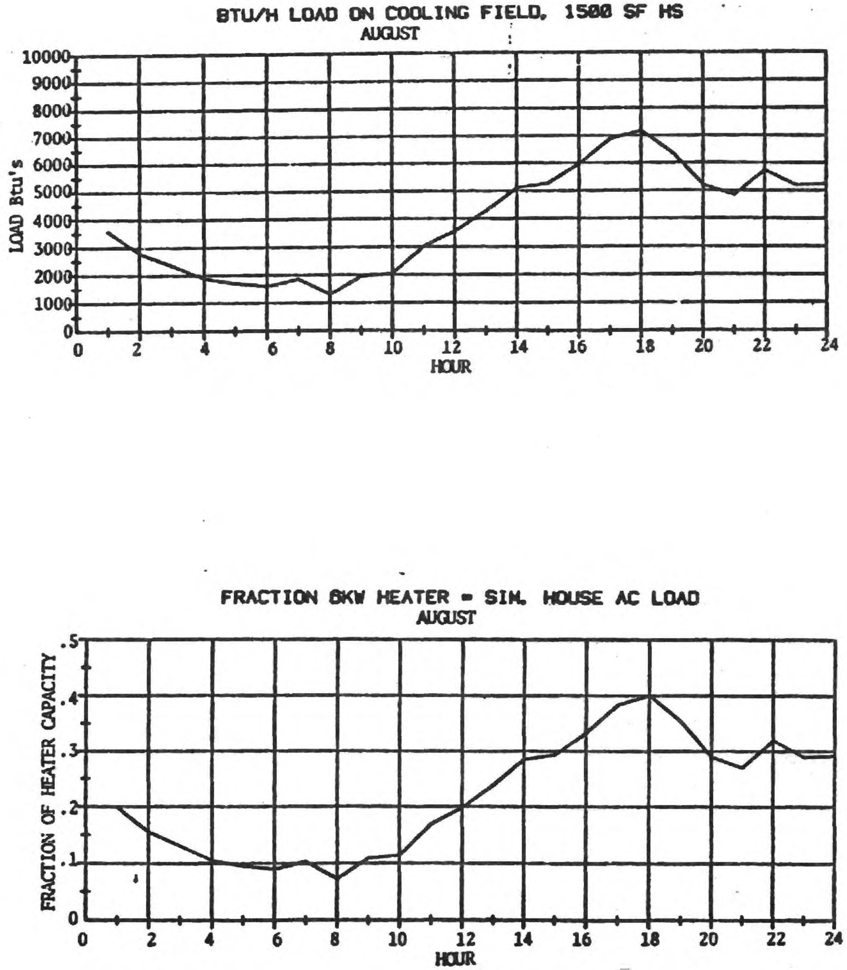
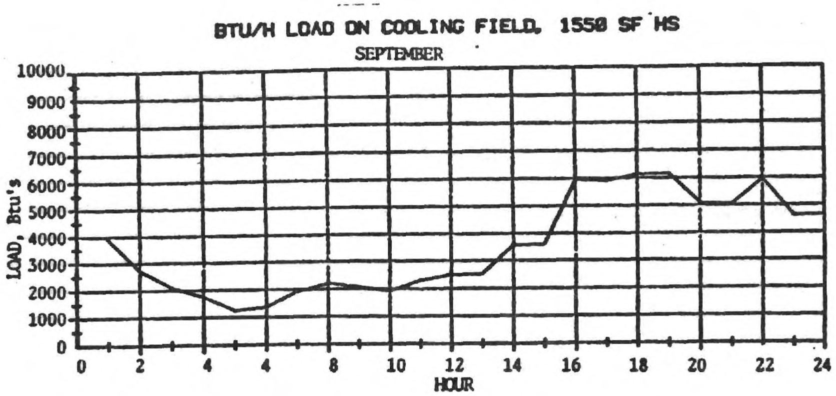


Figure 6.12 Building Load Profile - August



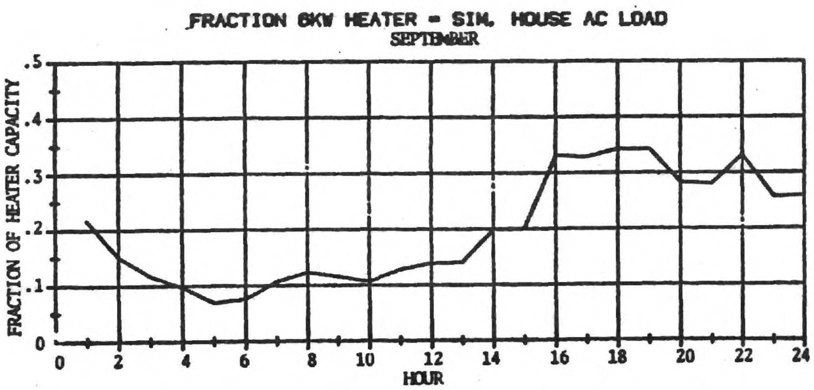


Figure 6.13 Building Load Profile - September

TABLE VI-II

Daily Sensible Cooling Load for Georgia Power Colombus House and Percentage Carried Passively

Month	Cooling Required (BTU/Month)	Cooling Passive (BTU/Month)	Percentage
May	1,280,216	0 *	0
June	2,508,555	1,421,515 **	56.7
July	2,444,049	1,024,920 ***	41.9
Aug.	2,158,381	0	0
Sept.	1,863,498	0	0
TOTAL	10,254,699	2,446,435	23.9

* Passive cooling not used in May due to load simulator failure

** Passive Cooling not started until 12 June due to load simulator failure

*** Cooling capacity of field exhausted on 16 July due to partial charging during previous winter week in January, losing three months when the field could have been charging. The soil temperature within the field at the start of charging was much higher than adjacent soil at that depth which had not been exposed to the installation procedure described above. Because of the high initial temperature, the insulation and the late start in charging, the field was not cooled to the temperature it would normally reach during a normal winter charging cycle.

Since we felt it undesirable to pump antifreeze solutions through the underground piping until we were positive we had control of all leaks, the system was operated with water throughout this winter. We had established that until the field temperature began to approach freezing, the water was unlikely to freeze if the pump kept the water circulating. It was also obvious that the water would freeze if the pump were to quit when the outside temperature was below freezing. The most probable cause of water not flowing was identified as pump stoppage caused by power interruptions. The pump and differential temperature control are therefore driven by a power inverter running off a battery. The battery is kept charged with a line-powered battery charger. This system keeps the pump running at all times when the ambient temperature is below freezing.

The field was scheduled to start carrying the simulated load in May 1981. Initial checkout showed all of the equipment functioning as desired, but when the load sequence was initiated in May, it failed to work properly. The problem was finally traced to a failure in a process controller. This controller is normally very reliable. The controller required off-site repair and was not available for use until the end of the second week in June.

Since the success of seasonal storage of cooling potential is highly dependent upon keeping the energy expended in charging the storage to a minimum, the field was initially charged using only natural connection over the heat exchanger. This proved satisfactory until the field temperatures dropped below 12.6 (55°F) at which point the charging rate became unacceptably low. Two 185-watt (631.4Btu/hr) 62.3m^3/m (2200 CFM) fans were added to the heat exchanger in late January to increase the charging rate. This modification increased the charging rate to acceptable levels although the fan capacity vs. power consumed vs. energy stored has not been optimized.

Since the cooling field performs like any sensible energy storage device with imperfect insulation, the field was losing cooling capability during the six week delay caused by the controller failure. Had the system been operable at the beginning of May, it would have passively carried all of the May and June cooling, despite the severe handicaps of the high initial temperatures and late initiation of charging. The system performance during this initial year of operation, despite all of the problems, leads one to be very optimistic about the potential of the DET concept.

Figure 6-14 shows the temperature distribution <u>across</u> the field at a 1.2m (4 ft.) depth for three different dates. One will notice the significant drop in field temperature between 27 January and 18 February and the decrease in temperature difference between the field and non-field is evident along either edge of the field. The increase in field temperature between 18 February and 24 February resulted from a very warm spell when the ambient temperature was above the field temperature and the charging system was not operating. The temperature increase resulted from migration of energy from the soil above and below four feet during this period.

Figure 6.15 shows the temperature distribution <u>along</u> the field center at the 1.2m (4ft.) depth for the same dates shown in Figure 6.14. The importance of the data shown in Figure 6.15 is the lack of significant temperature gradient from the field inlet end to the exit end.

2.2 Double Level Field

About the time that the decision had been made to abandon work on the single level field due to persistent leaks in the low desity polyethylene pipe Georgia Power offered to pay for complete replacement of the field. Georgia Power personnel had been following the progress of the program and are interested in seeing the tests completed.

Delays in getting corporate approval of the field replacement prevented replacing the field in early December 1981 as had been planned. Complete approval was not obtained until it was too late to get meaningful data during the winter of 1982. Additionally, unusually wet and cold weather prevented replacement of the field until early May 1982. Due to high interest in the concept by people all over the south, the field will be operated on a very low budget program so that data may be obtained on the field performance over a complete charge/discharge cycle.

Considerable soul searching and much effort has been directed toward determination of whether others might encounter similar leakage problems with the DET concept. Extensive conversation with Georgia Power personnel regarding their installation procedures for their underground power cables has led us to believe that the great quantities of both curbstone and fieldstone left from the old road provided an unusally harsh environment for our field. Georgia Power only cautions their installation personnel to not allow rocks to be put back over the cables for the first foot.

Similar talks with the Atlanta Gas Light Company about their installation procedures for plastic underground gas lines indicates slightly greater care. Plastic gas line is made from a special formula medium density polythylene as opposed to the low density commercial grade polyethylene used in the single level field. The gas pipe has a wall thickness of 3.8mm (.150 in.) rather than the wall thickness of 3.29mm (.108 in.) used in the original field. Fortunately, the 2306 gas pipe is readily available and is only

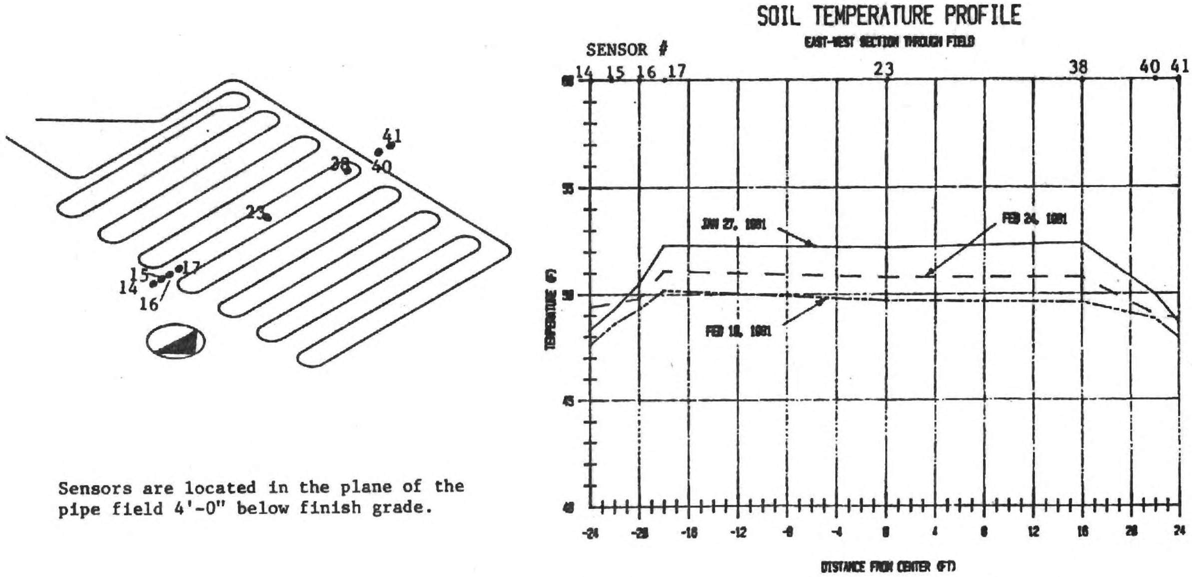


Figure 6.14 Temperature Distribution Across Cooling Field

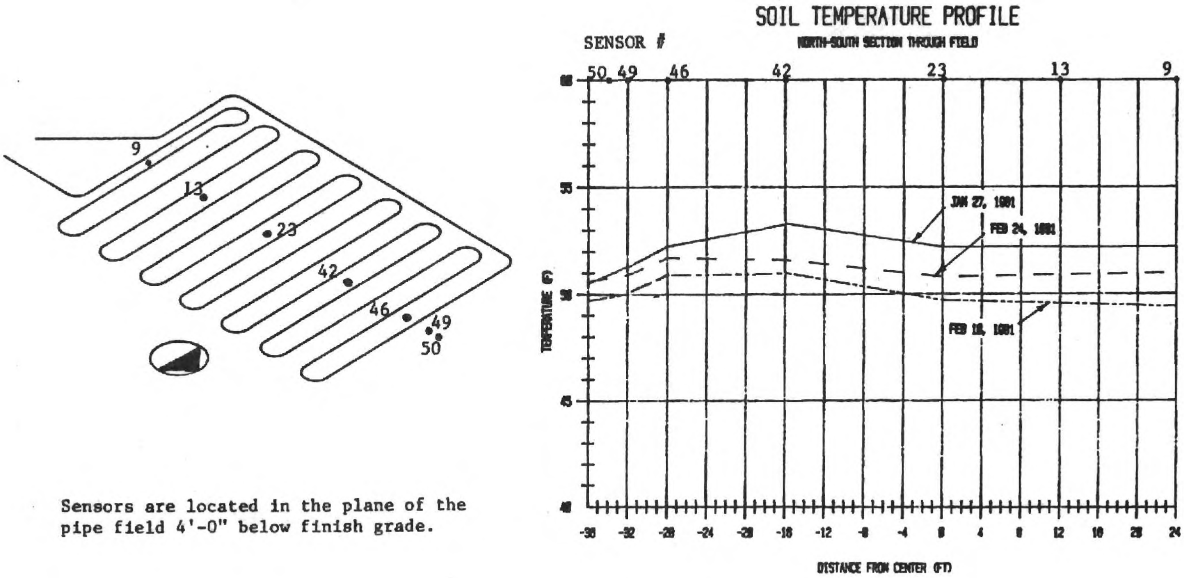


Figure 6.15 Temperature Distribution Along Cooling Field

slightly more expensive than the general purpose polyethylene pipe used in the original field. The plastic gas lines are also backfilled with screened soil for the first foot before soil which has been removed from the hole in which the pipe is installed is used.

Approximately 304.9m (1000 ft.) of 1250mil (1.25 in.) nominal inside diameter 2306 polyethylene gas pipe was used in the new double level field. The 1250mil (1.25 in.) diameter pipe was used because it is more readily available in the Atlanta area. The field was excavated to a depth of 2.6m (8.5 ft.), backfilled with .152m (6 in.) of screened general purpose sand before the lower pipe level was laid. Use of the .032m (1.25 in.) nominal inside diameter pipe permitted us to install the pipe on .9375m (3 ft.) centers rather than the 1.2m (4 ft.) centers used with the .04m (1.5 in.) pipe. Once the lower coils were inplace they were covered with an additional .15m (6 in.) of screened general purpose sand before the hole was backfilled with .93m (3 ft.) of soil removed from the hole. Extreme care was exercised to remove all rocks larger than .025m (1 in.) from the soil returned to the hole. Another .15m (6 in.) of screened general purpose sand was added before the top coil was laid. Once the top coil was in place a final .15m (6 in.) of screened sand was added as a cover before the hole was filled to .9375 (1 ft.) below grade. Again, extreme care was exercised to remove rocks from the backfill. The field was insulated with .05m (2 in.) of Dow SM extrude polystyrene insulation .9375m (1 ft.) below grade and covered with a .15mm (6 mil) plastic vapor barrier to prevent rain movement through the field. Finally, the last foot of soil was added, the site graded and seeded with bermuda grass seed.

The computer simulations had showed it was desirable to minimize plan area, i.e., reduce the area through which the field could gain energy from the surface. The pipe arrays are both within an area of 111.5m^2(4.14 x 12.2) (30'x40') with the lower coil 2.4m (8 ft.) below grade and the top coil 1.2m (4 ft.) below grade. The insulation was extended 2.4m (8 ft.) beyond the edge of the coils in all directions. Figure 6.16 shows

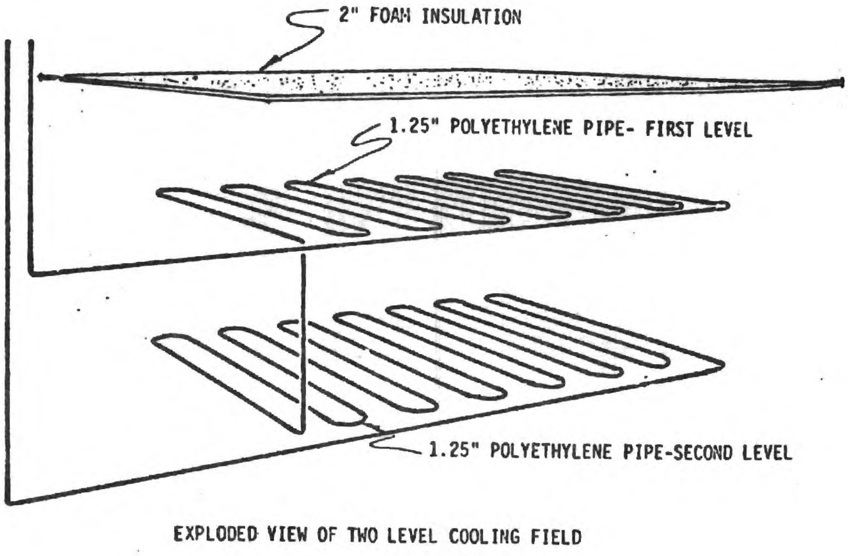


Figure 6.16 Exploded View of Double Level Cooling Field

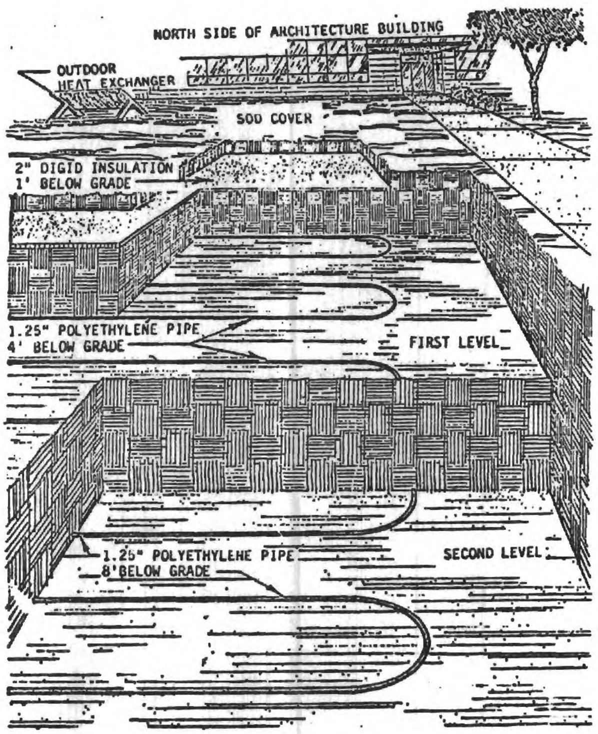


Figure 6.17 Cross Section of Double Level Cooling Field

an exploded view of the new field, while Figure 6.17 shows a cross section of the field as it was installed.

The pipe comes from the building, goes through a $1.67m^2$ (18 sq. ft.) air-water heat exchanger, goes directly into the ground to the 2.4m (8 ft.) level and runs to the far end of the field where it serpentines back toward the building until 152.4m (500 ft.) has been run. It then rises to the 1.2m (4 ft.) level where it runs to the far end of the field and serpentines back toward the building until approximately 152.4m (500 ft.) has been run. It then rises to the surface and runs directly back into the building where the pumps, and flow meters are located. With this arrangement, the surface area that was insulated was reduced by 40%, thus reducing one of the more costly items in the field. With less plan area, less energy will be lost to the surface. Additionally, the area is now sufficiently small to fit under most reasonably sized houses.

CHAPTER VII

DETACHED EARTH TEMPERING

RADIANT COOLING

1.0 INTRODUCTION

Earth coupled cooling requires a heat exchange element in the building that maintains its cooling efficacy at relatively high (22.7°C) (72.9°F) ground temperatures. Decoupling the earth from the building element and transferring heat from these interior surfaces to a cooling field with water, allows the occupant control over the temperature of the space and improves the cooling potential of the earth. Radiant wall panel heat exchangers were identified as viable options. Preliminary experiments with concrete slab cooling panels indicates that mass incorporated in a radiant element yields more uniform panel temperatures and maximizes cooling field potential. Radiant coupling to a secondary mass increases the response time of the slab and maintains cool temperatures in the secondary mass.

Passively cooling a building by using the ground as a heat sink has been popularly accepted in the form of underground, or "earth-coupled" structures. Directly coupling building elements such as the walls, floor, or ceiling to the earth has numerous disadvantages, among them the lack of control over the temperatures and heat transfer rates of the structural building elements. This lack of control leads to problems in hot-humid climates including condensation on walls in early spring, high wall tempertures in the fall and excessively cold surfaces in late winter. By decoupling the heat sink from the building element, control may be achieved over the building's interior temperature and the building element's heat flow to the earth.

The Detached Earth Tempering concept discussed in previous chapters comprises two elements: 1. an earth coupled element and 2. a building side heat-exchanger. Polyethylene pipe buried below the surface transfers heat from the building's interior to the ground. The block of earth associated with the pipe is separated from ambient temperatures by a two inch extruded polystyrene insulation one foot below grade. Temperatures in the test field have ranged from 14°C (58°F) in early spring to 22.7°C (72.9°F) in late July with simulated building cooling loads applied to the field. Field temperatures limit the type of heat exchangers practical in residences and limits the rate at which heat may be instantaneously removed from a building.

Earth coupled cooling is characterized by low temperature differences between the earth and the space that requires cooling. For example, in Atlanta earth temperatures at the end of August at a depth of 2.44m (8 ft.) are approximately 20.5°C (68.9°F). This temperature when directly coupled to building walls will increase due to the transfer of heat from the building to the earth. These relatively high cooling temperatures preclude the use of convective elements for maintaining comfortable air temperatures. Unless coupled to a mechanical cooling mechanism capable of producing a large temperature potential between the interior air and heat exchange fluid, convective heat exchanges, when coupled with earth heat sinks, play a secondary role to radiant heat exchange.

Radiant space conditioning panels are capable of maintaining comfort conditions with higher temperatures than forced air systems due to a reduction in mean radiant temperature caused by the cool panels[1]. In an extreme example, comfort may be maintained by a mean radiant temperature of 22.7°C (72.9°F) with a concurrent 33.9°C (93.0°F) air temperature, in a room where a person is sitting (1 MET) (72.9°F) wearing light clothes (0.9 CLO) while the surrounding air is moving at 0.1m/s and relative humidity is 20%. The example illustrates the effectiveness of space conditioning with radiant surfaces.

The above example has been overly simplified and the following issues must be considered before panel design is tested:

1. Mean radiant temperature is not the temperature of a radiant heat exchanger, but is rather the weighted sum of all surface temperatures in a room. The placement and distribution of radiant panels must maximize radiant exposure of the panel.

2. Surface temperatures of the panel can only be maintained at low temperatures with respect to air temperatures if they have an extremely high heat transfer rate. If dT=10°C (18°F), between panel surface and air, then the heat transfer fluid must transfer approximately $7.4W/m^2$ (2.35 Btu/hr Ft^2) from the panel to the cooling source. A second alternative to instantaneous heat transfer from the room to the fluid is to provide the panel with heat storage capacity.

Radiant heating systems were extensively used during the 1950's in residences, especially in cold climates. Copper pipe was typically laid in concrete slab floors and heated water used to keep the surface at a constant temperature. Noted for their comfort,[2] widespread application of the systems did not occur due to the relatively high installation costs when compared with forced air systems and to control problems in moderate climates. When used in 0.15m (.5 Ft.) thick floor slabs, response time to instantaneous loads were reported as a major drawback of the system. Typical cooling applications in commercial buildings used light weight steel panels in ceilings to reduce the response time of the mechancial cooling system. Water in these panels was mechanically cooled and because of the light weight and high conductance of the panels, the surfaces responded quickly to the peak cooling demands of commercial environments.

Although most radiant heat transfer surfaces have been located in floors and ceilings, both the floor and ceiling of a room present a smaller radiant surface than do walls to the sitting and standing postures of occupants. Although radiant cooling is not strictly similiar to radiant heating, analogies may be drawn between the two. For example, in a 4.9m x 4.9m (16.1 Ft x 16.1 Ft) room, under a 2.44m (8 Ft) high radiant ceiling, a maximum of 2.5% of the heat is transferred from the ceiling to the occupant in a radiant heating system. A floor would radiate only 2.25% of its energy to the subject[3]. A person standing in the middle of the room will receive 6% of the radiant energy from two opposing walls, or 9% of the energy if all four walls are effective radiant surfaces. In addition to the comfort considerations, walls are effective convective heat exchange surfaces due to the vertical attitude. Walls typically comprise the largest portion of the mean radiant surfaces and are the most effective building elements for interior heat exchange.

With tightly built, well insulated homes, both internal gains and the ambient climate combine to impose a cooling load on the air-conditioning system. A daily profile of the cooling load in a $158 m^2$ ($1700 Ft^2$) house with a 3808 W (12,997 Btu/hr) design heat loss (with a temperature differential of 11.1^oC (20^oC))is illustrated in Fig. 7.1. With this type of daily cooling load profile, the instantaneous loads on a radiant cooling panel coupled to an earth-coupled cooling source would overwhelm the system's cooling capacity. For example, radiant cooling walls deployed in a typical dwelling would have to remove up to 2636W (8997 Btu/hr) to the cooling field in order to maintain comfort.

Simulations indicate that the earth's instantaneous capacity would be quickly depleted with this type of instantaneous load and the ability to meet cooling demands substantially reduced. The cooling load must be more evenly spread over the diurnal cycle in order to increase the field's effectiveness.

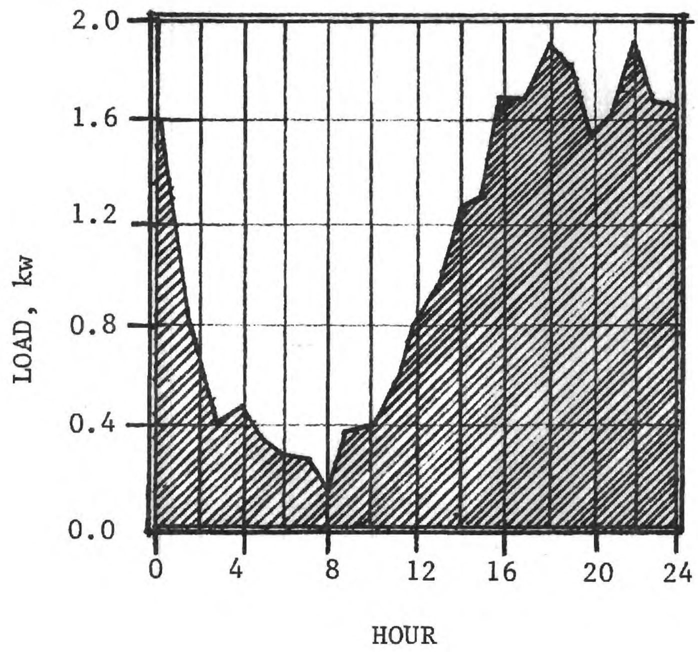


Figure 7.1 Diurnal Cooling Load for June -Georgia

To maintain comfort conditions using the ground's relatively high and slowly changing temperature, a concrete radiant panel system was designed and tested for application in residences as the interior heat exchange element for the earth-coupled cooling field.

2.0 RADIANT PANEL PERFORMANCE

Successful application of a radiant wall cooling system requires matching the available earth cooling potential with the panel's heat transfer characteristics.

Criteria for radiant concrete panels include heat exchange fluid (water) flow rate, slab thickness, pipe spacing within the wall, and coupling to other masses in the space. Application of high-mass radiant panels in a residence is not common and will require careful detailing before application. Poured concrete walls with embedded copper pipe are non-standard residential construction elements, but have been used for testing purposes in this study. The concrete slabs have .013m (.5 in) diameter copper pipe embedded on the rear of the slab.

Detailing of a practical residential radiant wall cooling system is seen as a future task once the proper performance specifications have been developed. For this purpose, a calorimetric box was constructed in the research space of the College of Architecture to measure the thermal performance of high-mass radiant panels.

In order to isolate the thermal performance of the slab and provide an adaptable interior space for testing various wall combinations, the calorimetric box was constructed with 0.1m moveable polyurethane walls. These walls define a constant 1.82 m^3 (64 Ft^3) volume (1.22 m x 1.22m x 1.22 m) (4 Ft x 4 Ft x 4 Ft) and allow the addition of brick or drywall materials to the interior cube surfaces. Edge losses are minimized by 0.25m x 0.25m (.82 Ft x .82 Ft) polyurethane members defining the edges of the cube. Conductance of the cube has been calculated at 1.46 W^{o}C (2.77 Btu/hr^{o}F). Baffled resistance lighting was used in the cube to impose a cooling load on the radiant panel being tested.

Radiant cooling panel varying from .062" thick steel to 4" thick concrete slabs have been evaluated in the calometric box. Typically the thickness of the slab being tested represents only half of a typical slab with both sides exposed. Losses from the rear panel of the slab are negligible. Thermcouples are distributed in the calorimetric box and slab as shown in the exploded view of the calorimetric box in Fig. 7.2. Heat flux sensors have also been applied to the front face of the slab.

2.1 Performance of Radiant Walls

During this initial series of experiments, three attributes of high-mass radiant wall systems were observed: 1. The surface temperatures of the slab are not influenced by a change in flow rates for pipe flow rates tested; 2. The surface temperature of the slab is influenced by radiant coupling to secondary mass in the space; and 3. Surface temperature distribution is uniform with diurnal cooling load profiles. This last attribute is crucial for maintaining comfort and heat transfer capability with wide pipe spacing and acceptable costs. The slabs were tested with their front facing the interior of the cube and its hardboard surfaces. A second test with each slab incorporated a solid core brick wall opposite the radiant cooling slab. Both steady state heat flux and a diurnal load simulating the potential gains on the radiant cooling panel were tested.

Steady state performance of a .05m (2 in.) thick slab with copper pipes spaced at .25m (10 in) intervals on the back of the slab indicated the response time of the slab to cooling loads. A continuous load of 354W (1161 Btu/hr) was generated in the interior of the cube. With water flowing through the slab, 60% of final equilibrium slab surface temperature was reached in 4.5 hours. A total of 13 hours was required to achieve steady state. Temperature profiles of the air and slab surface are illustrated in Fig. 7.3 Distribution of temperatures through sections of the 0.05m (2 in) slab are illustrated in Fig. 7.4. A 5.5 C° (9.9°F) temperature differential was observed between the slab surface over the pipe, and the slab surface between the pipes. Air temperatures were 19.4°C (34.9°F) higher than surface temperatures over the pipe. Temperature gradients within the slab sections are less pronounced at the midpoints between pipes than at pipe sections. Because applied radiant wall elements will rarely be subject to steady state loads, intermittent cooling loads are a more realistic test of the performance of radiant elements. The imposed cooling loads illustrate late evening peaks and morning lows and are similiar to those illustrated in Fig. 7.1.

Heat applied to the slab was weighted by a ratio of area of the test slab to the example application. This accounted for 1.6% of the buildings total radiant wall areas. This heat flux was doubled to account for spatial concentration of loads within a

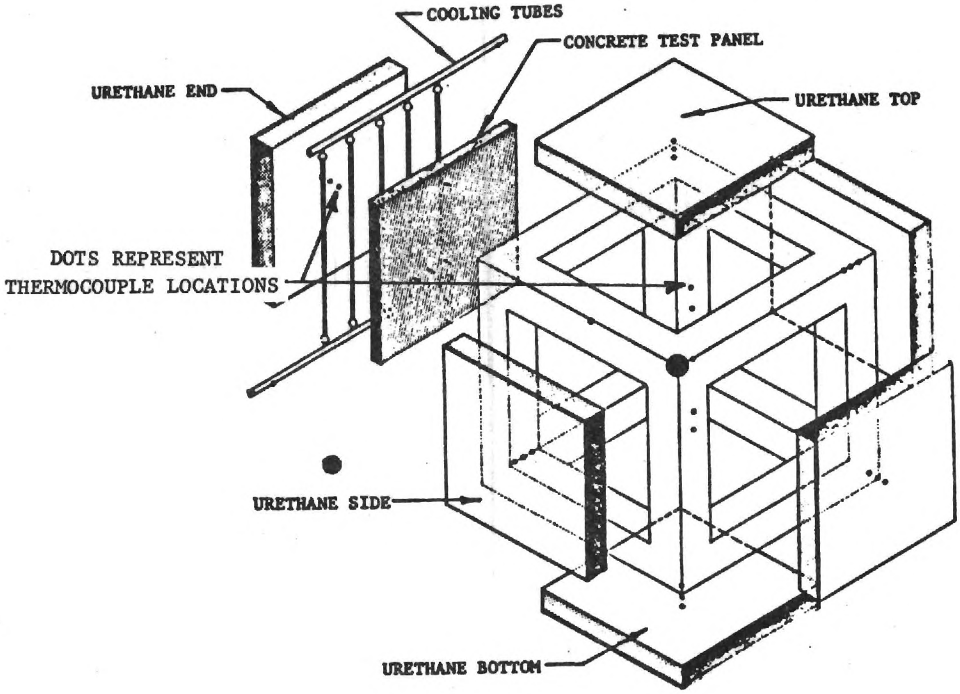


Figure 7.2 Exploded View of Calorimetric Test Box

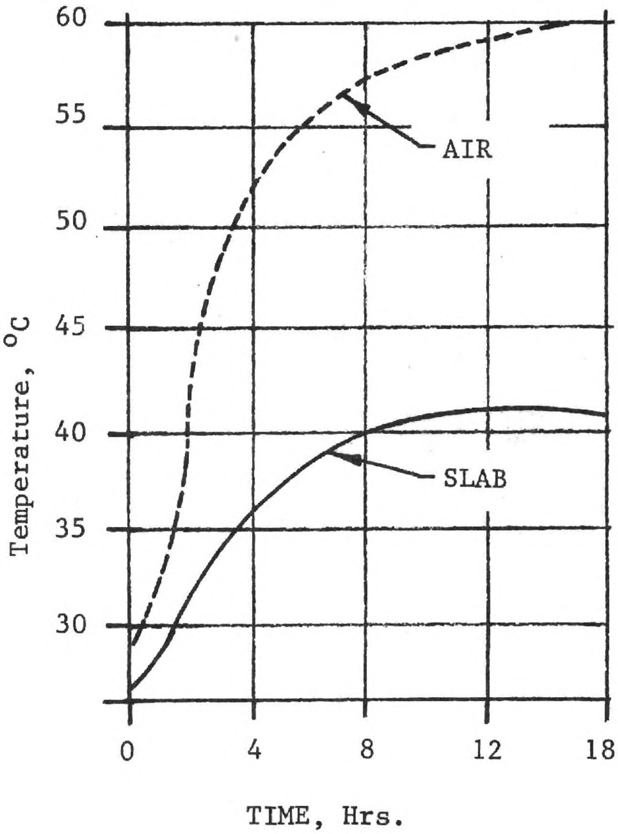


Figure 7.3 Steady State Temperature Profile in Calorimetric Box

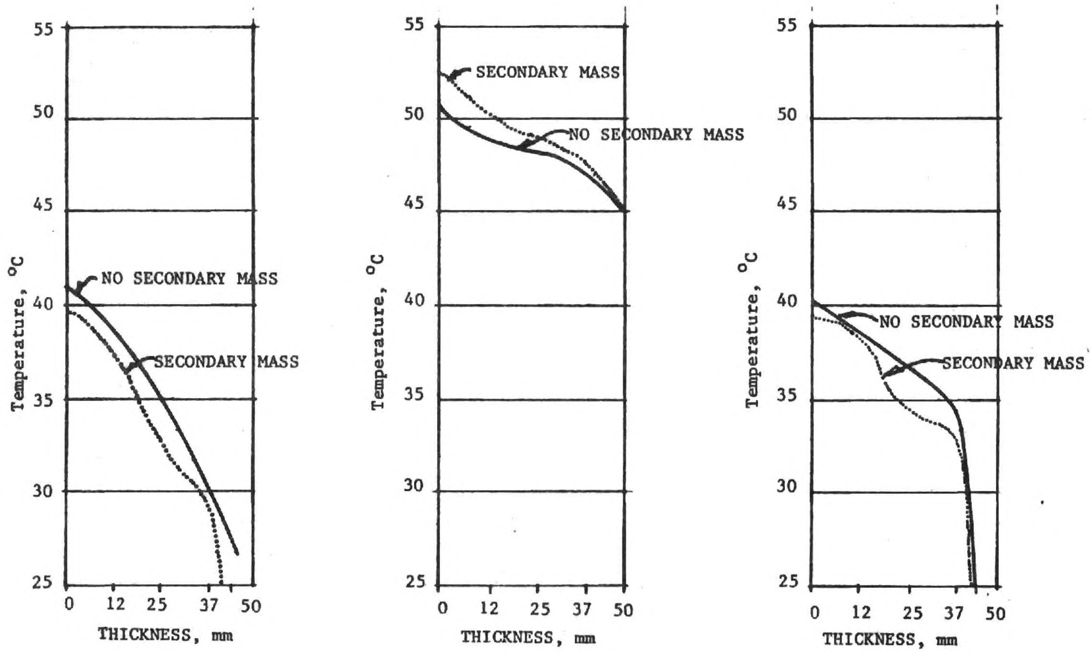


Figure 7.4 Temperature Distributions Through the Slab

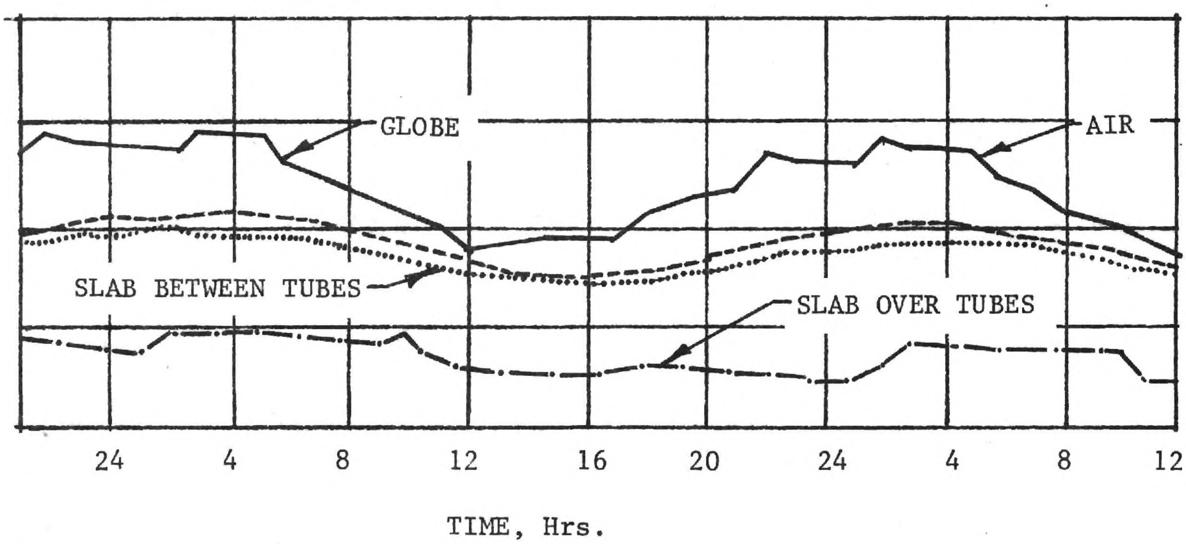


Figure 7.5 Temperatures in the Calorimetric Box Under Diurnal Loads

residence. The panel's surface temperatures follow air temperatures throughout the cooling load cycle as shown in Fig. 7.5. Simulations of the radiant panel performance and actual measurements indicate the uniformity of the surface temperatures on the slab. A 1.38°C (2.5°F) difference was recorded at times of maximum heat flux. Comparison with steady state performance illustrates the interdependence of the imposed cooling load profile, the radiant wall panel design and the radiant wall surface temperatures. Sections of the slab, shown in Fig. 7.6 illustrate the temperature gradient through the slab with a varying diurnal heat input. With this profile heat input to the box, heat flux to the slab varied from 8.52W/m^{2}C (1.5 Btu/hr°F) during peak heat input to 7.38W/m^{2}C (1.29 Btu/hr°F) at the low point in the morning. These values have been calculated from the temperature measurements and simulations of slab performance on a finite difference model. Surface temperature of the slab under these

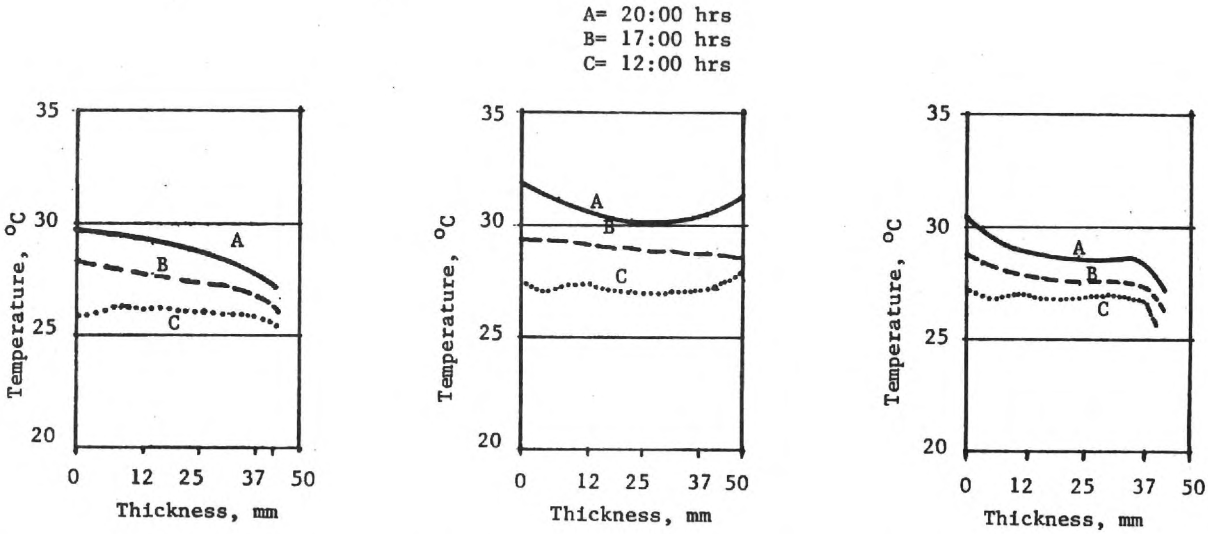


Figure 7.6 Temperature Gradient Through Slab with Diurnal Heat Input

conditions were typically 1.4°C (2.5°F) above the average water temperature. During peak hours this difference increases to 3.3°C (5.9°F). The surface to ambient temperature difference at these times is 4.4°C (7.9°F). This is well within comfort conditions assuming supply water is 22.8°C (73.0°F) rather than 26.6°C as (79.9°F) indicated in the tests.

Addition of the brick wall opposite the slab influenced the time period required for the test assembly to reach steady state. Temperatures of both the air and slab at steady state were not significantly different from the steady state condition without the brick wall, as shown in Fig. 7.4. Heat flux sensors mounted on the slab for these tests, did not agree with each other. Measurements at the surface over the pipe indicated lower heat flux (8.52W/m/m^2C) (1.5 Btu/hr°F) than at the midpoints between pipes (14.7W/m^2C) (2.59 Btu/hr°F). A third sensor was used to measure the convective, radiative split at a point on the surface over a pipe. If compared to measurements taken at the surface over the central pipe, the convection and radiation coefficients were 4.05 W/m^2C (.71 Btu/hr°F) and 4.26 W/m^2C (.75 Btu/hr°F) respectively.

Only a qualitative indication of the damping influence of the brick wall has been determined. Brick wall temperatures were recorded 0.6°C (1.1°F) below air temperatures indicating depression of the secondary mass' temperature by radiation to the cooling panel.

Considerable testing has been directed toward determining the radiative cooling potential of numerous different radiant cooling test panels using the radiant panel test box. Figures 7.7-7.11 summarize the performance of five different test panel configurations. One can see the improvement in performance under a transient load environment when mass is added to the radiant panel. All of the panels have been tested under both steady state and transient load conditions. Since steady state temperature are only dependent upon the inside-to-outside temperature and the thermal

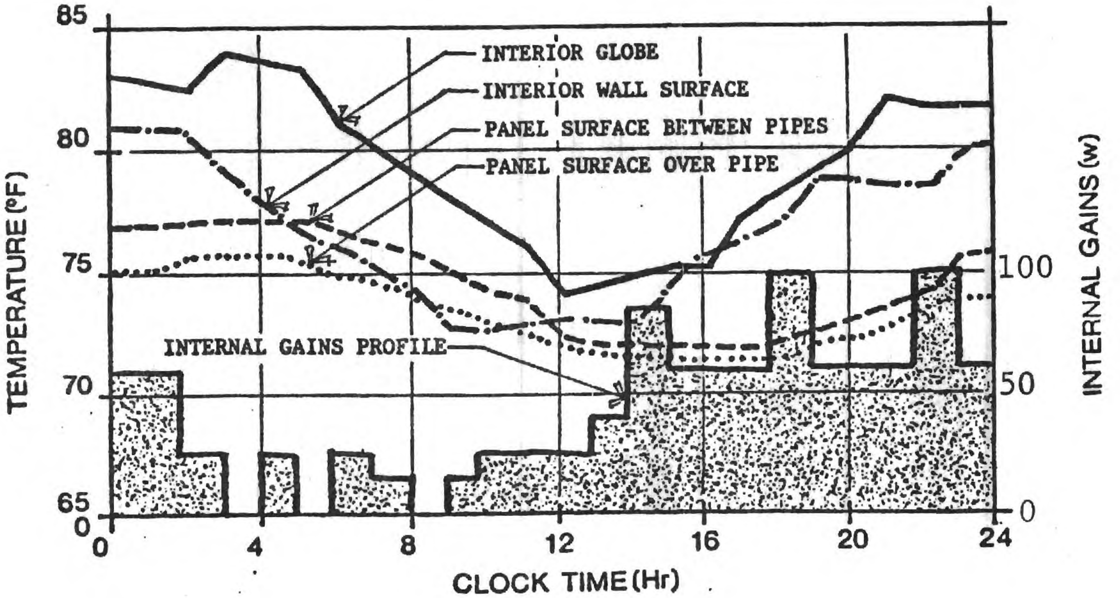


Figure 7.7 Thermal Performance of 2" (Half Thickness) Radiant Cooling Panel with 10" Tube Spacing

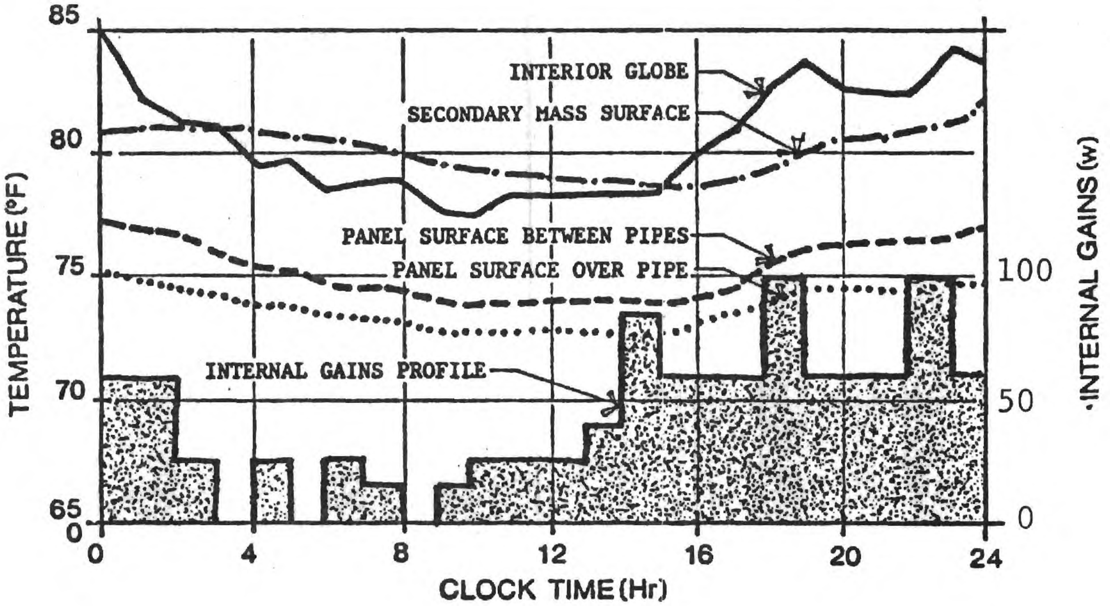


Figure 7.8 Thermal Performance of 2" Radiant Cooling Panel with Secondary Mass Wall, 10" Tube Spacing

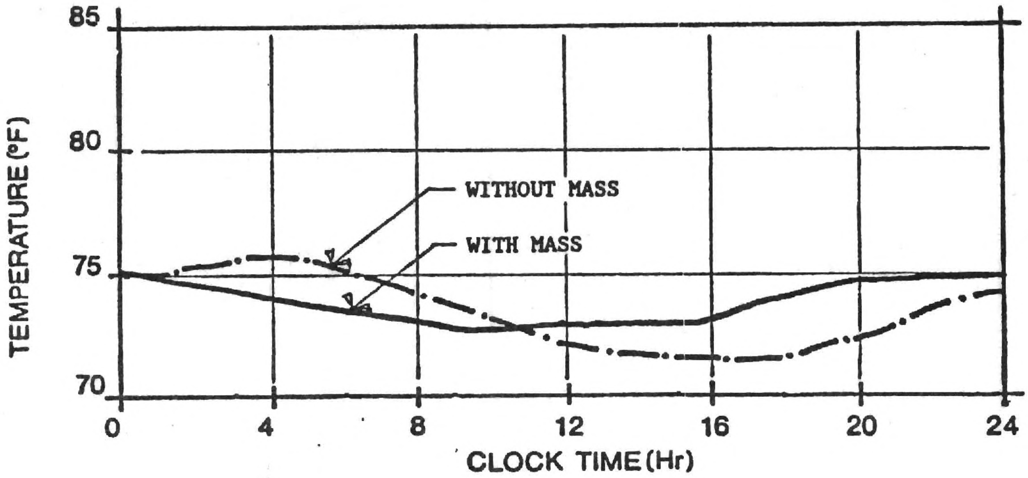


Figure 7.9 Panel Surface Temperature Directly Over Tubes when Subjected to a Dynamic Load, 10" Tube Spacing

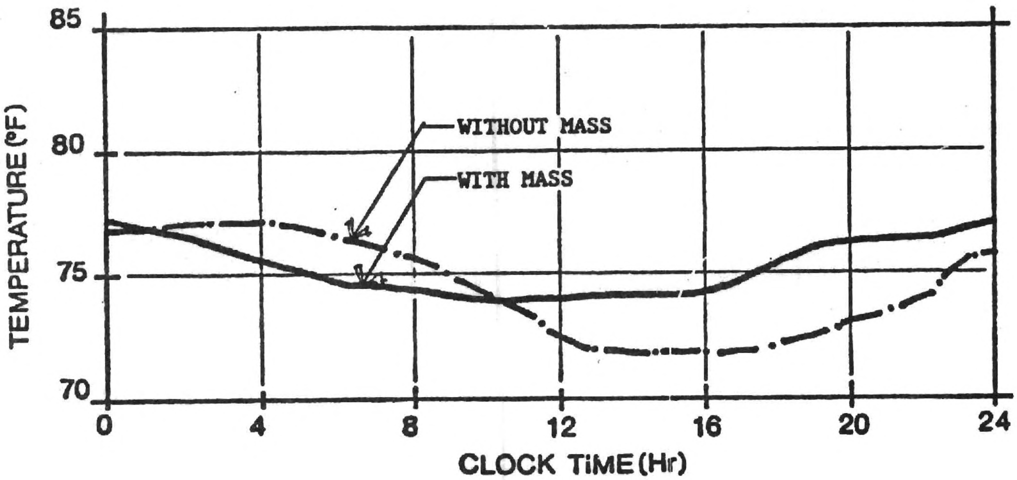


Figure 7.10 Panel Surface Temperature Midway Between Tubes when Subjected to a Dynamic Load, 10" Tube Spacing

resistance of the walls, we will concentrate here on the transient load analysis.

Figure 7.7 shows the load profile used in all of the tests. Figure 7.7 also shows the variation in temperature of four different locations for a .05m (2 in) half thickness wall with water tubes spaced .25m (10 in) apart. Figure 7.8 shows the same test except with a brick wall added to the test box directly opposite the radiant cooling wall. Notice that all of the temperatures fluctuate much less when the additional mass is added. Figures 7.9 and 7.10 further illustrate the importance of secondary mass on the performance of the radiative cooling panel. Figure 7.9 shows how the additional mass decreases the temperature fluctuations of the panel directly over the tubes, while Figure 7.10 shows a similar effect midway between the tubes.

It had been suggested by several people interested in this program that performance of the system could be greatly improved by using a radiant panel with a high conductivity, similar to the metal panels used in radiant ceiling heating systems. Figure 7.11 shows this assumption of improved performance with increased conductivity is not valid if one must sacrifice mass and conductive area for increased conductivity. The steel panel used in the tests whose results are shown in Figure 7.11 was .0016m (.062 in) thick with copper tubes bonded with high conductivity epoxy every .25m (10 in) (the same tubes and spacing used with the concrete panels whose performance is shown in Figures 7.7-7.10). With water at a fixed temperature and flow rate, the steel panel is not able to conduct energy to the area directly over the tubes fast enough to meet the instantaneous loads. With very little inherrent mass the steel panel can not store cooling capacity as do the concrete panels. One can't disagree with the premise that high conductivity is desirable. One can disagree if one must sacrifice mass to obtain high conductivity. Since the field studies have already shown that the field can probably meet the average daily load, but might be incapable of meeting the instantaneous peak loads these tests are reassuring.

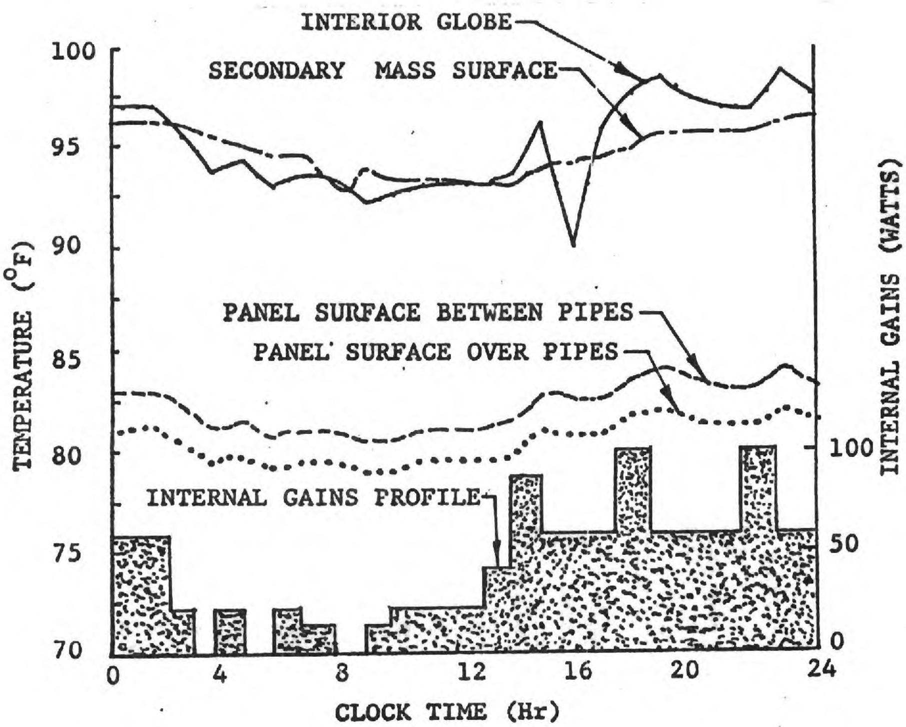


Figure 7.11 Thermal Performance of a .062" Thick Steel Radiant Cooling Panel with 10" Tube Spacing

3.0 RADIANT COOLING

The success of the detached earth tempering concept is dependent upon radiatively coupling the cooling potential to the house rather than through a water-to-air heat exchanger as is conventionally done in residences. It has been pointed out that radiant heating has been less than successful in the southern part of the United States because the climate changes so rapidly that the system is required to go from cooling to heating more rapidly than is possible with massive radiant elements. It should be pointed out that the lack of success of radiant heating in this area is not due to the radiant element but to the poor design and construction of the remainder of the house. Radiant heating or cooling with massive radiant elements will obviously suffer from very poor control if the UA of the building is high. High building UA's require that the radiant element operate at temperatures significantly different from the interior air temperature. Notice that the radiant floor in the example discussed above is always within 1.5F of the air temperature and is within the comfort zone. It will provide heating or cooling with only a 3 F change in air temperature should the weather change suddenly.

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CHAPTER VIII

LATENT LOADS

There are no passive cooling techniques which will remove latent loads in hot-humid climates. Although it may be possible to develop dehumidification techniques which appear passive, it is unlikely they will be as simple, efficient or cost effective as more conventional dehumidification techniques.

1.0 LOAD MINIMIZATION

It is usually necessary to use an electrically driven dehumidifier to reduce latent loads in hot humid climates. This is not a passive approach although it might be considered passive if a dehumidifier such as that marketed by E-Tech[1] is employed. This device was designed as a heat pump powered domestic hot water heater and as such has been shown to use 50-70% less energy than a conventional electric hot water heater. If the evaporator of the heat pump hot water heater is located within the structure, it dehumidifies and to some degree cools the air while using the energy it removed from the air to heat the domestic hot water. If the energy used to drive the heat pump hot water heater is allocated as water heating energy, the dehumidification and slight cooling is a no-energy side benefit.

Rather than trying to devise methods to passively handle a latent load, a better approach might be to design the system to control the latent loads. It was stated earlier that proper load control was a first consideration in all passive designs. Just as proper use of overhangs, color, windows location, orientation, etc., are effective and necessary in limiting sensible loads; control of infiltration and ventilation is effective and necessary in limiting latent loads. Since internal latent loads are similar everywhere, the primary difference in latent loads in structures is that resulting from infiltration and ventilation. If all infiltration and ventilation were eliminated, latent loads would be the same regardless of the climate. Obviously, one can't eliminate infiltration. It is generally agreed that 1/3 to 1/2 of the building's volume per hour in fresh air (infiltration and/or ventilation) is necessary to minimize health problems, reduce odors, etc. Most passive structure have 1-1.5 volume air changes each hour. One sees that through proper design one can reduce latent loads by a factors of 3 to 5 just by designing the building to have less infiltration. Further reductions are possible through the use of an enthalphy exchanger and positive infiltration. Houses in Northern Canada and Alaska are working very satisfactorily with air changes less than .05 other than that supplied through the enthalpy exchanger.

An enthalpy exchanger is essentially a device much like a heat exchanger which is capable of the transfer of both energy and moisture. If cool dry inside air is exhausted through the enthalpy exchanger while hot-humid outside air is being pulled inside through the enthalpy exchanger the two air streams exchange both energy and moisture. Readily available exchangers are capable of reaching efficiencies of 80% or greater. This means the makeup air is within 80% of the conditions of the inside air. This gives both a sensible and latent ventilation load reduction of 80%. By combining the enthalpy exchanger with reduced infiltration rates one can reduce latent loads due to external ambient conditions by a factor of 12 to 20 compared to a typical convential or typical passive structure.

As one can see from the brief discussion above, passive structures can be designed for hot-humid climates which have latent loads very similar to the latent loads for structures in dry climates. This permits the passive cooling effort to be directed primarily toward sensible loads. This approach requires passively cooled structures designed for hot-humid climates to be very tight, i.e., have very low infiltration rates.

At the same time, it is highly desirable to open up structures during the fall and spring to take advantage of moderate ambient temperatures and humidities during these times. Since the enthalpy exchanger works just as well during the heating season, the structure needs to again be tightly closed during the winter months.

2.0 RUN-AROUND CYCLE

One encounters conditions in passively cooled buildings where it is desirable to lower the dewpoint temperature without lowering the dry-bulb temperature. Unfortunately, conventional mechanical dehumidifers are very inefficient (always operating at a COP less than one). Figure 8.1 shows the component layout and basic operating mode of conventional dehumidifiers. Notice that all the energy removed by the evaporator is added back to the air at the condenser, including the latent energy resulting from the water removal and the work required to move the energy.

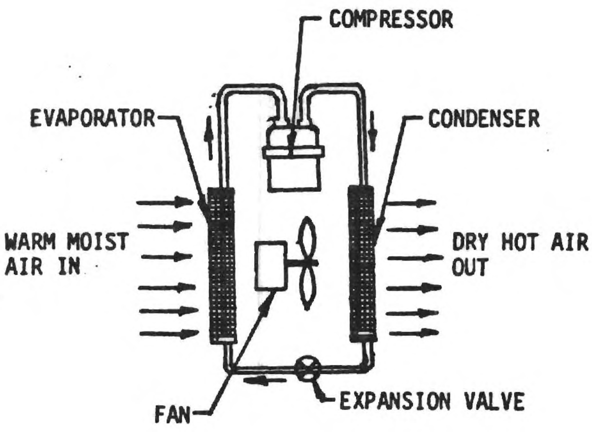

Figure 8.1 Basic Components of a Conventional Dehumidifier

Conventional air or water source heat pumps are capable of much higher COP's. Figure 8.2 shows the component layout and basic operating mode of a water source heat pump using well water as a source and a discharge well as receiver. A water source heat pump will provide both sensible and latent cooling which in some cases may be undesirable.

In those cases where dehumidification is the desired product one must modify the conventional heat pump cycle. The run-around cycle or the addition of run-around coils to a new system or to an existing system is one very good way to increase system latent capacity without increasing the system total capacity. This cycle has the advantage of low operating cost, compared to other methods of increasing latent capacity.

Figure 8.3 shows the basic run-around cycle. Water is circulated between two water-to-water heat exchangers, one located on each side of the mechanical system's cooling coil. Sensible heat withdrawn from the warm air on its way to the cooling coil is carried by the circulating water to the reheat coil. The reheat coil then returns the sensible heat to the chilled air leaving the cooling coil. Any sensible heat added to the flowing air by the reheat oil is exactly equal to the heat removed by the precooling coil. Consequently, there is a decrease in the refrigeration required to reach a given dewpoint temperature when using the run-around cycle.

Figure 8.4 shows the conventional dehumidifier cycle drawn on a psychrometric chart, while Figure 8.5 shows the heat pump cycle with a run-around coil added. Notice that the final air temperature with the conventional dehumidifier is higher than the initial temperature although the moisture level has been lowered. Figure 8.5 shows that the run-around cycle when added to the conventional heat pump results in dehumidification with some sensible cooling. Point "A" in Figure 8.5 is the final air condition for the heat pump without the run-around coil.

Passively cooled buildings frequently are able to carry the sensible load but are

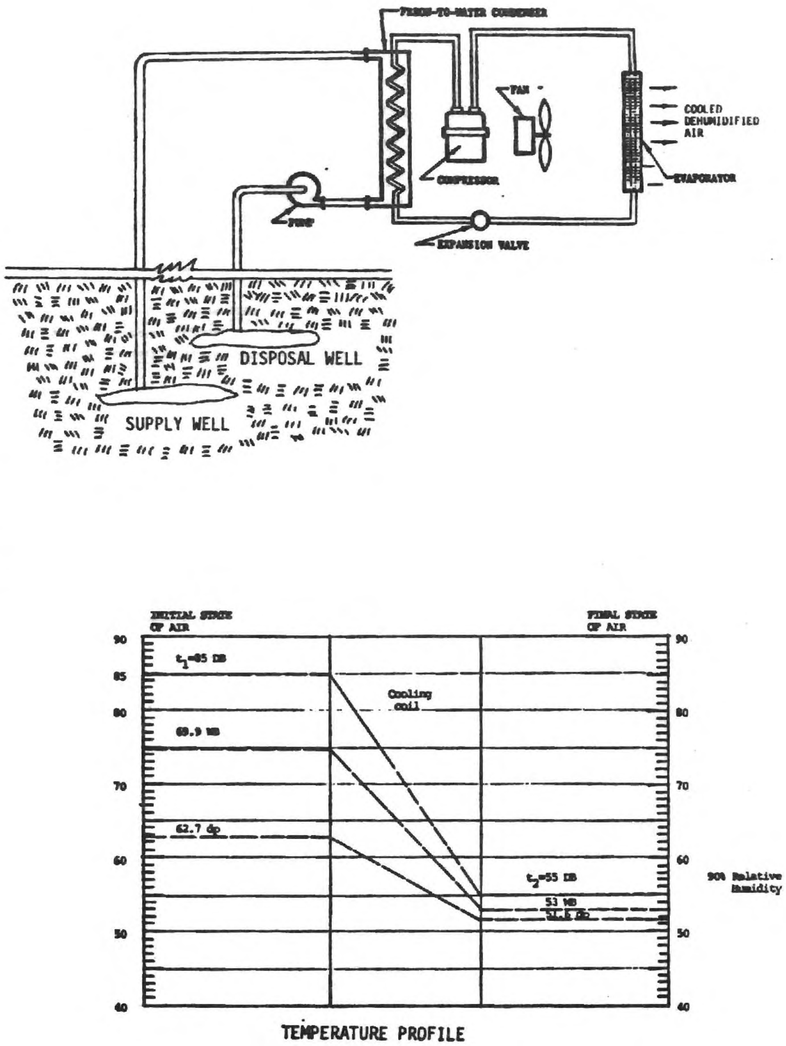

Figure 8.2 Water Source Heat Pump

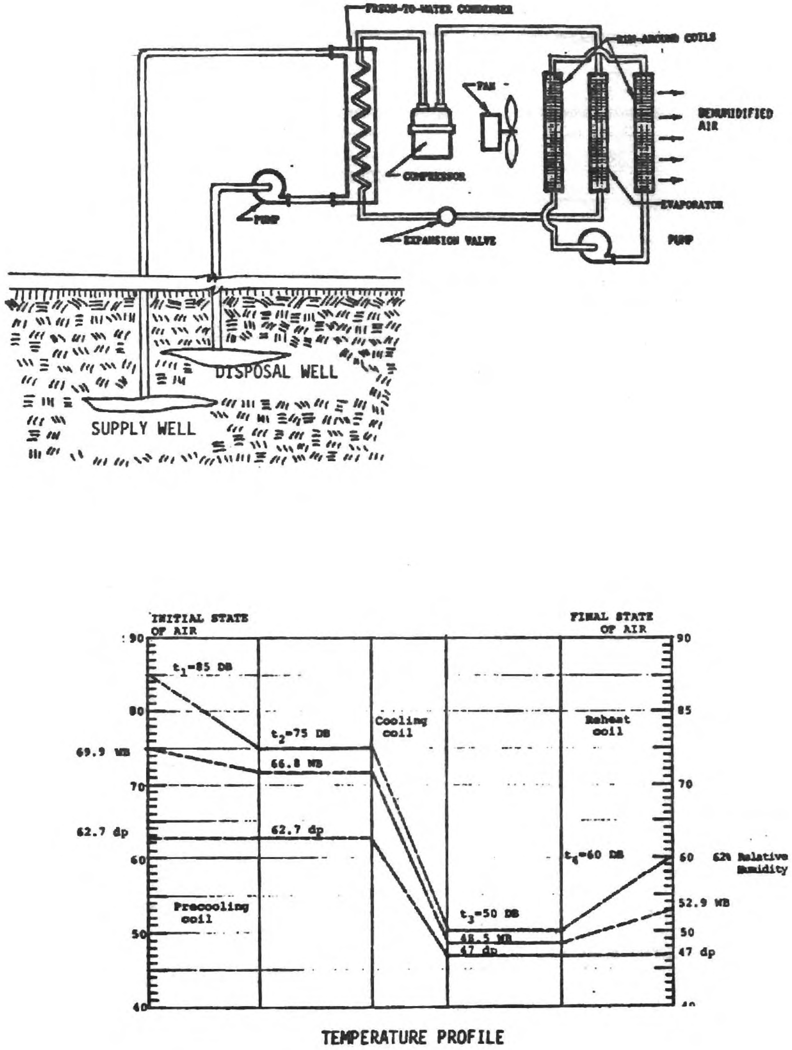

Figure 8.3 Water source Heat Pump with Run-Around Coil

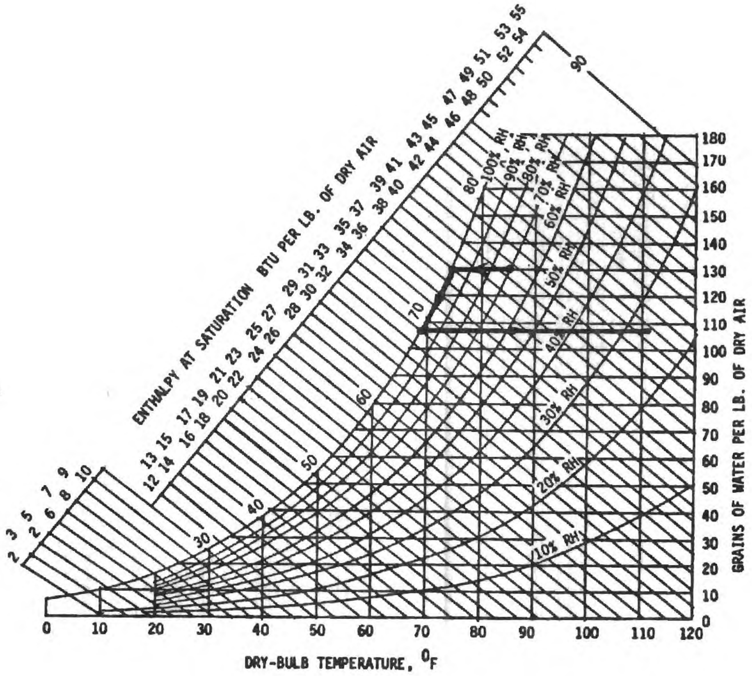

Figure 8.4 Dehumidification Using Conventional Dehumidifier

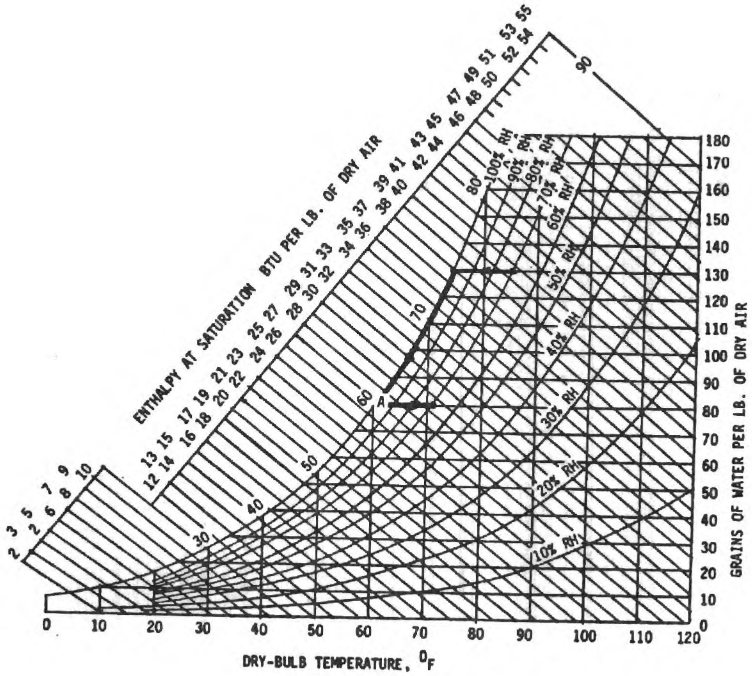

Figure 8.5 Conventional Heat Pump with Run-Around Coil

unable to meet the latent loads. If conventional mechanical systems are added to these buildings, the mechancial system will carry both sensible and latent loads. Sensible loads should not be carried by mechanical equipment if they can be carried passively. Addition of a run-around coil will shift the mechanical system toward more latent cooling and less sensible cooling, thus improving system efficiency.

3.0 HEAT PUMP DOMESTIC HOT WATER HEATER

If one succeeds in totally eliminating heating and cooling loads in a residence, one finds that a substantial utility load still remains. It is not uncommon for domestic hot water (DHW) energy requirements to exceed the heating and cooling requirements for well designed energy efficient residences.

Solar DHW heaters are one solution to this problem, but one usually finds solar DHW installation costs to run from $2500-3500. Recently several manufacturers have begun to market DHW heaters which operate on the heat pump principle. These heat pump DHW heaters require only 35-50% as much energy input as required by conventional electric DHW heaters and can be purchased for $550-650. Since active solar DHW heaters become quite expensive when designed to deliver more than 60% of the total energy requirements, the heat pump DHW units are able to deliver comparable percentages at unit costs less than 30% of solar units.

Since the heat pump DHW units are very small and easily moved, it is possible to locate the units within occupied spaces. This makes it possible for the unit to not only meet the DHW needs, they can also provide sensible and latent cooling as a side benefit. Figure 8.6 shows the basic design of a heat pump DHW heater. One will notice that the unit is essentially a small water source heat pump which takes the energy removed from the space being cooled and adds this energy to the domestic hot water. This is very similar to the energy reclaim units which are added to conventional heat pumps.

As discussed earlier it is desirable to provide sensible cooling using passive means. If cooling can be accomplished passively, mechanical sensible cooling, even if it is a by product of a domestic hot water heating unit, decreases the load that can be carried by the passive system. This suggest that it is desirable to decrease the heat pump DHW unit's sensible cooling capacity and increase its latent cooling capacity. Figure 8.7 shows a schematic of a unit which has been modified through the addition of a run-around coil as dicussed in above. The run-around coil increases the latent capacity and decreases the sensible cooling capacity without significantly affecting the efficiency of the system.

This now permits one to heat their domestic hot water efficiently, dehumidify (latent cool), and still provide the sensible cooling passively. If one looks at the combined efficiency of this system, one finds that the system COP is now about 5.0.

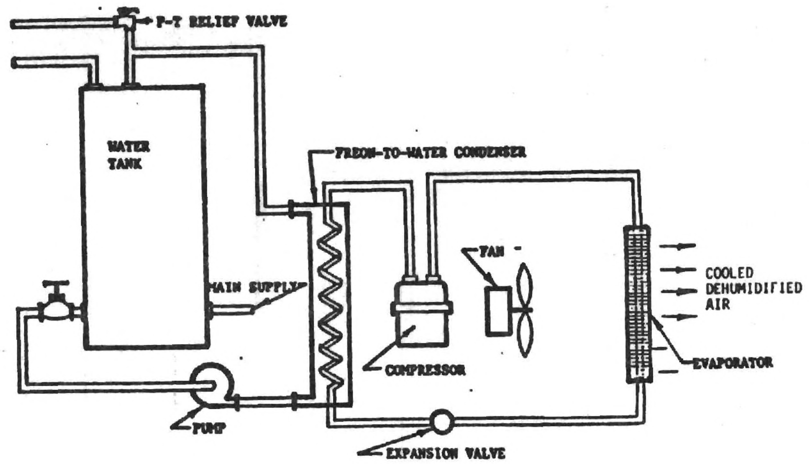

Figure 8.6 Conventional Heat Pump DHW Heater

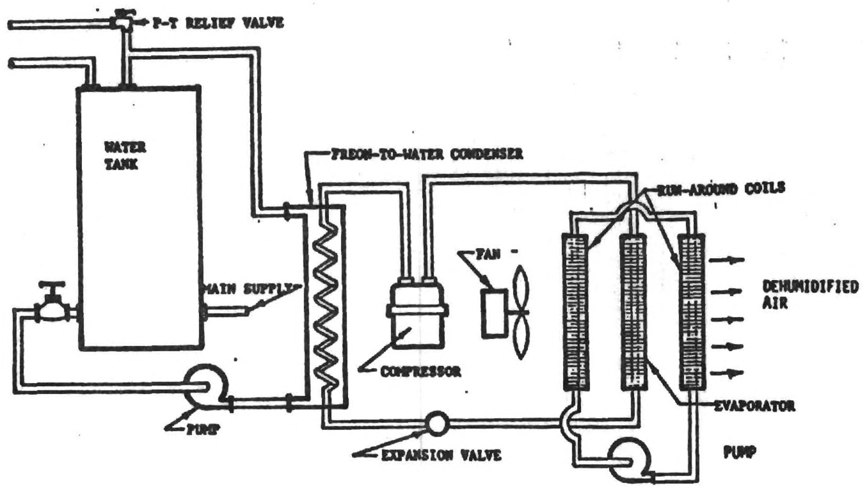

Figure 8.7 Heat Pump DHW Heater with Run-Around Coil

REFERENCES

1. "Heat Pump Water Heater", Technical Literature from E-Tech, Inc., 3570 American Drive, Atlanta, Georgia.

CHAPTER IX

CONCLUSIONS AND RECOMMENDATIONS

While the experimental program suffered some difficulties, the Detached Earth Tempering concept appears to have considerable potential as a passive cooling concept for hot-humid climates. The problems experienced were not a result of inherrent problems with the concept, but resulted from problems in implementing the concept. Program scheduling resulted in the field being installed during the record heat wave being experienced in Atlanta during the summer of 1980. This resulted in the ground insulated from the surface being at a much higher temperature than it would normally be at that depth and time of the year. The late installation also resulted in the field getting only a marginal charge during the first winter. Instrumentation equipment failure also resulted in six weeks of lost cooling capacity during May and early June.

The program did identify problems related to the installation method used for the field. Failure to remove large rocks from the field backfill resulted in leakage at a number of places in the field. The difficulty experienced in locating and repairing the field leaks emphasizes the great need to install a cooling field in a manner that will eliminate leaks. The second field was constructed using much better materials and an installation method that eliminated the rocks which had led to the leaks in the original field.

Computer simulation of the original field showed that field performance could be greatly improved by going to a double level field with a reduced plan area. The second field was installed using the lessons learned with the original field and the results from the computer simulations.

These simulations also showed that placing the field beneath the house would not only improve field performance but may also be capable of cooling the residence directly through the floor. This particular design needs considerably more computer simulation before an installation is made. Installation of the field beneath the house without considerable additional work directed toward optimizing the concept could result in a house which could potentially overcool the residence during the early summer.

RECOMMENDATIONS

It is recommended that the second field be charged and discharged over a complete year so that performance data can be developed for system operation over a complete year without problems. The new field has been installed and the instrumentation put in place.

It is also recommended that one or more of the new water walls currently becoming available be installed in the radiant cooling facility and extensively evaluated. These walls should have considerably improved performance over the concrete walls evaluated thus far in the program. These water walls have similar mass, greater energy storage capacity, and much higher thermal conductivity. This should result in a better capabiltiy to meet transient loads.

It is recommended that considerable additional simulation effort be directed toward optimizing and quantifying the performance of the double level field located beneath the house. This particular concept promises much improved performance with little additional cost. Cost may actually be reduced with the field beneath the house due to reduced insulation costs.

It is recommended that the runaround coil on the DHW heatpump be evaluated extensively and effort be directed toward optimizing the coil for latent cooling.

APPENDICES

A. GROCS/TRNSYS INPUT
B. GROCS/TRNSYS OUTPUT

APPENDIX A

SAMPLE OF GROCS INPUT

```
*
*      10TH RUN ON "FIELD" - NO INSULATION -UNDER HOUSE
*    SIMULATION BASED ON GROCS MODEL NO.1
*
     SIMULATION  0  2000  1.
     TOLERANCES     .002  .002
     LIMITS          50    50
     WIDTH           132
*
*
            CONSTANTS 3
                   MAXM = 4000.
                   Q    = 6000.
                   TIN  =   72.
*
*
*
                   UNIT 1  TYPE 15       CONTROLLER
                   PARAMETERS 11
                               0 0 0 0 4 0 -7 2 4 8 4
                   INPUTS 5
                               0,0 0,0 0,0 4,1 0,0
                               MAXM MAXM TIN 55. Q
*
*
            UNIT 2  TYPE 15      Q SUM OPERATOR
                   PARAMETERS 31
                 0 0 3 0 3 0 3 0 3 0 3 0 3 0 3 0 3 0 3 0 3 0 3 0 3 0 3 0 3 0 3
                   INPUTS 16
      4,3 4,4 4,5 4,6 4,7 4,8 4,9 4,10 4,11 4,12 4,13 4,14 4,15 4,16 4,17 4,18
          0   0   0   0   0   0   0   0    0    0    0    0    0    0    0    0
*
            UNIT 3  TYPE 15      Q CALC OPERATOR
                   PARAMETERS 5
                               0 0 0 4 1
                   INPUTS 3
                               4,2 0,0 4,1
                               0   TIN 0
*
           UNIT 4  TYPE 29           PIPE COIL FROM BNL
           PARAMETERS 65
                1.0
  10  10.    80.1   .5
  11  10.    80.1   .5
  12  10.    80.1   .5
  13  10.    80.1   .5
  14  10.    80.1   .5
  15  10.    80.1   .5
  16   10.    80.1    .5
  17   10.    80.1    .5
  22  10.    80.1   .5
  23  10.    80.1   .5
  24  10.    80.1   .5
  25  10.    80.1   .5
  26  10.    80.1   .5
  27  10.    80.1   .5
  28   10.    80.1   .5
  29   10.    80.1   .5
           INPUTS 2
                0,0 1,1
                               TIN 10000.
           DERIVATIVES 1
                0
*
           UNIT 5  TYPE 25           PRINTER
```

```
          PARAMETERS 4
              200. 0. 2000. 0.
          INPUTS 4
           4,1 2,1 3,1 4,2
           TOUT QSUM QCALC MDOT
           UNIT 6 TYPE 26       PLOTTER
           PARAMETERS 4
           20. 0. 2000. 0.
           INPUTS 4
               4,1 2,1 3,1 4,2
               TOUT QSUM QCALC MDOT
           END
GROCS
          7
   0.0     0.5     2.0     4.5     8.5     14.0     19.0
   44.     46.     50.     56.     62.     65.      65.
   43.     44.     47.     52.     58.     63.      64.
   48.     48.     49.     52.     56.     61.      63.
   56.     56.     54.     54.     56.     60.      62.
   67.     65.     62.     59.     58.     59.      61.
   76.     74.     70.     65.     61.     60.      61.
   82.     80.     76.     70.     64.     61.      61.
   83.     82.     79.     74.     68.     63.      62.
   78.     78.     77.     74.     70.     65.      63.
   70.     70.     72.     72.     70.     66.      64.
   59.     61.     64.     67.     68.     67.      65.
   50.     52.     56.     61.     65.     66.      65.
         1.        5
      0.82
        40      7
  1     65.    0.    13200.    .5
  2     65.    0.      907.5   28.
  3     65.    .5     2576.   35.
  4     55.    3.0     3440.     35.
  5     55.    2.0      600.     35.
  6     55.    2.0      600.     35.
  7     55.    2.0      600.     35.
  8     55.    2.0      600.     35.
  9     55.    3.0     3440.     35.
  10     55.   3.5       300.     35.
  11    55.    4.5      300.     35.
  12   55.     3.5     300.     35.
  13   55.     4.5     300.     35.
  14   55.     3.5     300.     35.
  15   55.     4.5     300.     35.
  16   55.     3.5     300.     35.
  17    55.    4.5      300.     35.
  18    55.    6.0      600.     35.
  19    55.    6.0      600.     35.
  20    55.    6.0      600.     35.
  21    55.    6.0      600.     35.
  22    55.    7.5      300.     35.
  23   55.     8.5     300.     35.
  24    55.    7.5      300.     35.
  25    55.    8.5      300.     35.
  26    55.    7.5      300.     35.
  27    55.    8.5      300.0    35.
  28    55.    7.5     300.0     35.
  29    55.    8.5     300.0     35.
  30    55.      8.5    3440.0     35.
  31    55.      8.5    3440.0     35.
  32       55.    10.0      600.0     35.
  33       55.    10.0      600.0     35.
  34       55.    10.0      600.0     35.
  35       55.    10.0      600.      35.
  36       55.    14.0      4128.0    35.
```

```
37      55.     12.0      1200.0    35.
38      55.     12.0      1200.0    35.
39      55.     14.0      4128.0    35.
40      55.     15.0      4800.0    35.
1     0.0
2      0.5
3      2.0
4      4.5
5      8.5
6     14.0
7     19.0
113
1   2   8077.   3.0
1  41   1141.   3.0
2   3   2804.   1.25
3   4   688.  3.0
3   5    300.    1.5
3   6    300.     1.5
3   7    300.     1.5
3   8    300.    1.5
3   9    688.    3.0
3   42   204.   102.0
4   5    100.    13.95
4   6     40.    19.0
4   9     80.    45.64
4    10     50.    13.95
4  11   50.0   13.95
4  12  20.0   19.0
4  13  20.0   19.0
4  18  50.0   13.95
4  19  20.0   19.0
4  31  688.0  5.0
4   43    204.     8.0
4   44    306.     8.0
5   6     60.    10.0
5  10    300.     1.5
6   7     60.    10.0
6    12   300.     1.5
7   8     60.    10.0
7   9     40.    19.0
7  14    300.0  1.5
8  9     100.0  13.95
8  16    300.0  1.5
9  16     50.0  13.95
9   17    50.    13.95
9   21    50.    13.95
9   30   688.     5.0
9   43   208.     8.0
9   44   306.     8.0
10  11    300.0    1.0
10  12    30.0   10.0
11  13    30.0   10.0
11   18    300.0    1.5
12  13   300.0   1.0
12  14    30.0  10.0
13  15    30.0  10.0
13  19   300.0   1.5
14  15   300.0   1.0
14  16    30.0  10.0
15  17    30.0  10.0
15  20   300.0   1.5
16  17   300.0   1.0
17  21    300.     1.5
18  19     60.    10.0
18  22    300.     1.5
18  31     50.    13.95
```

19	20	30.	10.0
19	24	300.0	1.5
19	31	20.0	19.0
20	21	30.0	10.0
20	26	300.0	1.5
20	30	20.0	19.0
21	28	300.0	1.5
21	30	50.0	13.95
22	23	300.0	1.0
22	24	30.0	10.0
22	31	50.0	13.95
23	25	30.0	10.0
23	31	50.0	13.95
23	32	300.0	1.5
24	25	300.0	1.0
24	26	30.0	10.0
24	31	20.0	19.0
25	27	30.0	10.0
25	31	20.0	19.0
25	33	300.0	1.5
26	27	300.0	1.0
26	28	30.0	10.0
26	30	20.0	19.0
27	29	30.0	10.0
27	30	20.0	19.0
27	34	300.0	1.5
28	29	300.0	1.0
28	30	50.0	13.95
29	30	50.0	13.95
29	35	300.0	1.5
30	31	80.0	45.64
30	34	40.0	19.0
30	35	100.0	13.95
30	39	688.0	5.5
30	45	510.0	8.0
31	32	100.0	13.95
31	33	40.0	19.0
31	36	688.0	5.5
31	45	510.0	8.0
32	33	60.	10.0
32	37	300.	2.0
33	34	60.	10.0
33	37	300.	2.0
34	35	60.	10.0
34	38	300.	2.0
35	38	300.	2.0
36	37	140.	16.33
36	39	96.	45.64
36	40	280.	22.82
36	46	612.	8.0
36	47	588.	6.0
37	38	60.	20.0
37	40	600.	10.0
38	39	140.	16.33
38	40	600.	10.0
39	40	280.	22.82
39	46	96.	45.64
39	47	688.	6.0
40	47	1200.	4.0
	200.		

EOI. 0 FILES. 1 RECS. 858 WORDS.

APPENDIX B

SAMPLE OF GROCS/TRNSYS OUTPUT

```
BIGTRN,FIELD10.
/
IDLE
```

```
1
              TRNSYS - A TRANSIENT SIMULATION PROGRAM
       FROM THE SOLAR ENERGY LAB AT THE UNIVERSITY OF WISCONSIN
                       VERSION  10.1  6/1/79


*     10TH RUN ON "FIELD" - NO INSULATION -UNDER HOUSE
*   SIMULATION BASED ON GROCS MODEL NO.1
*

 SIMULATION        0.          2.000E+03    1.000E+00

 TOLERANCES        2.000E-03   2.000E-03

 LIMITS   50   50
```

```
 WIDTH   132
*
*

 CONSTANTS  3
      MAX =  4.000E+03       Q   =  6.000E+03       TIN =  7.200E+01
*
*
*

 UNIT  1     TYPE 15          CONTROLLER
     PARAMETERS 11
     0.          0.          0.          0.          4.000E+00   0.          -7.000E+00  2.000E+00   4.000E+00
     8.000E+00   4.000E+00
     INPUTS  5
      0, 0        0, 0        0, 0        4, 1        0, 0
     4.000E+03   4.000E+03   7.200E+01   5.500E+01   6.000E+03
*
*

 UNIT  2     TYPE 15          Q SUM OPERATOR
     PARAMETERS 31
     0.          0.          3.000E+00   0.          3.000E+00   0.          3.000E+00   0.          3.000E+00
     0.          3.000E+00   0.          3.000E+00   0.          3.000E+00   0.          3.000E+00   0.
     3.000E+00   0.          3.000E+00   0.          3.000E+00   0.          3.000E+00   0.          3.000E+00
     0.          3.000E+00   0.          3.000E+00
     INPUTS 16
      4, 3        4, 4        4, 5        4, 6        4, 7        4, 8        4, 9        4,10        4,11
      4,12        4,13        4,14        4,15        4,16        4,17        4,18
     0.          0.          0.          0.          0.          0.          0.          0.          0.
     0.          0.          0.          0.          0.          0.          0.
*

 UNIT  3     TYPE 15          Q CALC OPERATOR
     PARAMETERS  5
     0.          0.          0.          4.000E+00   1.000E+00
     INPUTS  3
      4, 2        0, 0        4, 1
     0.          7.200E+01   0.
*

 UNIT  4     TYPE 29               PIPE COIL FROM BNL
     PARAMETERS 65
     1.000E+00   1.000E+01   1.000E+01   8.010E+01   5.000E-01   1.100E+01   1.000E+01   8.010E+01   5.000E-01
     1.200E+01   1.000E+01   8.010E+01   5.000E-01   1.300E+01   1.000E+01   8.010E+01   5.000E-01   1.400E+01
     1.000E+01   8.010E+01   5.000E-01   1.500E+01   1.000E+01   8.010E+01   5.000E-01   1.600E+01   1.000E+01
     8.010E+01   5.000E-01   1.700E+01   1.000E+01   8.010E+01   5.000E-01   2.200E+01   1.000E+01   8.010E+01
     5.000E-01   2.300E+01   1.000E+01   8.010E+01   5.000E-01   2.400E+01   1.000E+01   8.010E+01   5.000E-01
     2.500E+01   1.000E+01   8.010E+01   5.000E-01   2.600E+01   1.000E+01   8.010E+01   5.000E-01   2.700E+01
     1.000E+01   8.010E+01   5.000E-01   2.800E+01   1.000E+01   8.010E+01   5.000E-01   2.900E+01   1.000E+01
     8.010E+01   5.000E-01
     INPUTS  2
      0, 0        1, 1
     7.200E+01   1.000E+04
     DERIVATIVES  1
     0.
*

 UNIT  5     TYPE 25               PRINTER
     PARAMETERS  4
     2.000E+02   0.          2.000E+03   0.
     INPUTS  4
      4, 1        2, 1        3, 1        4, 2
     TOUT        QSUM        QCALC       MDOT
```

```
UNIT  6     TYPE 25          PLOTTER
     PARAMETERS  4
     2.000E+01    0.          2.000E+03    0.
     INPUTS  4
      4, 1         2, 1         3, 1         4, 2
     TOUT         QSUM         QCALC        MDOT

END


       TRANSIENT SIMULATION     STARTING AT TIME =  0.
                                STOPPING AT TIME =  2.000E+03
                                        TIMESTEP =     1
          DIFFERENTIAL EQUATION ERROR TOLERANCE =  2.000E-03
               ALGEBRAIC CONVERGENCE TOLERANCE =  2.000E-03
1
*     10TH RUN ON "FIELD" - NO INSULATION -UNDER HOUSE
*   SIMULATION BASED ON GROCS MODEL NO.1
*


*** THIS IS GROCS SPEAKING
                                          THE INITIAL TIME

HOUR=      1.00            MONTH=          5          TIME INTERVAL=      1.00

THERMAL CONDUCTIVITY =        .82
NO. FREE BLOCKS=    40 NO. RIGGED BLOCKS=     7
```

BLOCK NUMBER	DEPTH OF CENTER	VOLUME	VOLUME HEAT CAPACITY
1	0.00	13200.00	.50
2	0.00	987.50	28.00
3	.50	2576.00	35.00
4	3.00	3440.00	35.00
5	2.00	600.00	35.00
6	2.00	600.00	35.00
7	2.00	600.00	35.00
8	2.00	600.00	35.00
9	3.00	3440.00	35.00
10	3.50	300.00	35.00
11	4.50	300.00	35.00
12	3.50	300.00	35.00
13	4.50	300.00	35.00
14	3.50	300.00	35.00
15	4.50	300.00	35.00
16	3.50	300.00	35.00
17	4.50	300.00	35.00
18	6.00	600.00	35.00
19	6.00	600.00	35.00
20	6.00	600.00	35.00
21	6.00	600.00	35.00
22	7.50	300.00	35.00
23	8.50	300.00	35.00
24	7.50	300.00	35.00
25	8.50	300.00	35.00
26	7.50	300.00	35.00
27	8.50	300.00	35.00
28	7.50	300.00	35.00
29	8.50	300.00	35.00
30	8.50	3440.00	35.00
31	8.50	3440.00	35.00
32	10.00	600.00	35.00
33	10.00	600.00	35.00
34	10.00	600.00	35.00
35	10.00	600.00	35.00
36	14.00	4128.00	35.00

37	12.00	1200.00	35.00
38	12.00	1200.00	35.00
39	14.00	4128.00	35.00
40	15.00	4800.00	35.00
1	0.00		
2	.50		
3	2.00		
4	4.50		
5	8.50		
6	14.00		
7	19.00		
1	2	8077.00	3.00
1	41	1141.00	3.00
2	3	2804.00	1.25
3	4	688.00	3.00
3	5	300.00	1.50
3	6	300.00	1.50
3	7	300.00	1.50
3	8	300.00	1.50
3	9	688.00	3.00
3	42	204.00	102.00
4	5	100.00	13.95
4	6	40.00	19.00
4	9	80.00	45.64
4	10	50.00	13.95
4	11	50.00	13.95
4	12	20.00	19.00
4	13	20.00	19.00
4	18	50.00	13.95
4	19	20.00	19.00
4	31	688.00	5.00
4	43	204.00	8.00
4	44	306.00	8.00
5	6	60.00	10.00
5	10	300.00	1.50
6	7	60.00	10.00
6	12	300.00	1.50
7	8	60.00	10.00
7	9	40.00	19.00
7	14	300.00	1.50
8	9	100.00	13.95
8	16	300.00	1.50
9	16	50.00	13.95
9	17	50.00	13.95
9	21	50.00	13.95
9	30	688.00	5.00
9	43	208.00	8.00
9	44	306.00	8.00
10	11	300.00	1.00
10	12	30.00	10.00
11	13	30.00	10.00
11	18	300.00	1.50
12	13	300.00	1.00
12	14	30.00	10.00
13	15	30.00	10.00
13	19	300.00	1.50
14	15	300.00	1.00
14	16	30.00	10.00
15	17	30.00	10.00
15	20	300.00	1.50
16	17	300.00	1.00
17	21	300.00	1.50
18	19	60.00	10.00
18	22	300.00	1.50
18	31	50.00	13.95
19	20	30.00	10.00

19	24	300.00	1.50
19	31	20.00	19.00
20	21	30.00	10.00
20	26	300.00	1.50
20	30	20.00	19.00
21	28	300.00	1.50
21	30	50.00	13.95
22	23	300.00	1.00
22	24	30.00	10.00
22	31	50.00	13.95
23	25	30.00	10.00
23	31	50.00	13.95
23	32	300.00	1.50
24	25	300.00	1.00
24	26	30.00	10.00
24	31	20.00	19.00
25	27	30.00	10.00
25	31	20.00	19.00
25	33	300.00	1.50
26	27	300.00	1.00
26	28	30.00	10.00
26	30	20.00	19.00
27	29	30.00	10.00
27	30	20.00	19.00
27	34	300.00	1.50
28	29	300.00	1.00
28	30	50.00	13.95
29	30	50.00	13.95
29	35	300.00	1.50
30	31	80.00	45.64
30	34	40.00	19.00
30	35	100.00	13.95
30	39	688.00	5.50
30	45	510.00	8.00
31	32	100.00	13.95
31	33	40.00	19.00
31	36	688.00	5.50
31	45	510.00	8.00
32	33	60.00	10.00
32	37	300.00	2.00
33	34	60.00	10.00
33	37	300.00	2.00
34	35	60.00	10.00
34	38	300.00	2.00
35	38	300.00	2.00
36	37	140.00	16.33
36	39	96.00	45.64
36	40	280.00	22.82
36	46	612.00	8.00
36	47	588.00	6.00
37	38	60.00	20.00
37	40	600.00	10.00
38	39	140.00	16.33
38	40	600.00	10.00
39	40	280.00	22.82
39	46	96.00	45.64
39	47	688.00	6.00
40	47	1200.00	4.00

TIME = 0.0000

TOUT	QSUM	QCALC	MDOT
5.503E+01	0.	0.	3.529E+02

*** THIS IS GROCS SPEAKING

HOUR= 200.00 MONTH = 5 YEAR = 0

60.84	60.33	59.74	56.66	61.28	60.31	59.57	59.04	56.60	64.38
64.07	62.23	61.84	60.59	60.14	59.38	58.90	59.93	58.54	57.50
56.80	58.76	58.01	57.74	57.21	56.95	56.58	56.39	56.12	55.38
55.42	56.15	55.83	55.56	55.40	55.94	55.41	55.25	55.75	56.52

HEAT INPUT TO FREE BLOCKS FROM GROUND COIL AND OR TANK

0.	0.	0.	0.	0.	0.	0.	0.	0.	199193.
208609.	129848.	140110.	84043.	94829.	53544.	64519.	0.	0.	0.
0.	37273.	46475.	26935.	32916.	19205.	23106.	13289.	15904.	0.
0.	0.	0.	0.	0.	0.	0.	0.	0.	0.

TOTAL HEAT INPUT= 1189790.

TIME = 200.0000

TOUT	QSUM	QCALC	MDOT
5.697E+01	5.965E+03	5.995E+03	3.968E+02

THIS IS GROCS SPEAKING

HOUR= 400.00 MONTH = 5 YEAR = 0

62.31	61.61	60.89	57.73	62.92	62.13	61.38	60.72	57.61	66.11
66.13	64.52	64.44	63.09	62.09	61.83	61.54	63.12	61.77	60.40
59.38	61.88	60.80	60.85	60.03	59.81	59.13	58.91	58.39	55.95
56.07	58.28	57.83	57.18	56.80	56.64	56.78	56.20	56.37	57.54

HEAT INPUT TO FREE BLOCKS FROM GROUND COIL AND OR TANK

0.	0.	0.	0.	0.	0.	0.	0.	0.	333558.
346107.	239053.	255226.	172923.	191984.	125200.	146073.	0.	0.	0.
0.	76654.	109833.	64949.	89390.	55304.	73619.	45533.	58935.	0.
0.	0.	0.	0.	0.	0.	0.	0.	0.	0.

TOTAL HEAT INPUT= 2384431.

TIME = 400.0000

TOUT	QSUM	QCALC	MDOT
5.992E+01	5.965E+03	5.990E+03	4.940E+02

THIS IS GROCS SPEAKING

HOUR= 600.00 MONTH = 5 YEAR = 0

64.14	63.35	62.54	58.92	64.35	63.82	63.23	62.62	58.75	67.25
67.41	66.21	66.26	65.13	65.07	64.05	63.88	65.12	64.13	62.98
61.84	64.23	63.25	63.41	62.56	62.42	61.64	61.48	60.84	56.66
56.84	60.40	60.02	59.17	58.70	57.24	58.55	57.67	56.91	58.33

HEAT INPUT TO FREE BLOCKS FROM GROUND COIL AND OR TANK

0.	0.	0.	0.	0.	0.	0.	0.	0.	446654.
456579.	337320.	353685.	260365.	282686.	203899.	230666.	0.	0.	0.
0.	119617.	179093.	106979.	153598.	99502.	137177.	89615.	118581.	0.
0.	0.	0.	0.	0.	0.	0.	0.	0.	0.

TOTAL HEAT INPUT= 3576016.

TIME = 600.0000

TOUT	QSUM	QCALC	MDOT
6.293E+01	5.978E+03	5.978E+03	6.585E+02

THIS IS GROCS SPEAKING

HOUR= 70.00 MONTH = 6 YEAR = 0

66.02	65.15	64.23	60.22	65.70	65.41	65.00	64.50	60.04	68.24
68.45	67.64	67.78	66.93	66.97	66.10	66.03	66.73	66.12	65.22
64.21	66.28	65.38	65.72	64.89	64.90	64.08	64.06	63.36	57.46
57.70	62.42	62.16	61.27	60.80	57.79	60.38	59.39	57.43	59.01

HEAT INPUT TO FREE BLOCKS FROM GROUND COIL AND OR TANK

0.	0.	0.	0.	0.	0.	0.	0.	0.	542512.

```
 547564.  424548.  438216.  342768.  365283.  283987.  313821.       0.       0.       0.
      0.  170622.  254915.  155659.  224483.  151730.  210663.  144376.  191202.       0.
      0.       0.       0.       0.       0.       0.       0.       0.       0.       0.

TOTAL HEAT INPUT=  4762342.


                   TIME =  800.0000
    TOUT        QSUM        QCALC       MDOT
 6.601E+01   5.917E+03   5.946E+03   9.883E+02


*** THIS IS GROCS SPEAKING
HOUR=     270.00               MONTH =       6             YEAR =       0
   67.89    66.94    65.92    61.59    67.01    66.91    66.67    66.30    61.41    69.13
   69.36    68.89    69.09    68.52    68.64    67.97    68.01    68.16    67.89    67.30
   66.50    68.12    67.32    67.85    67.00    67.29    66.48    66.65    65.92    58.35
   58.61    64.32    64.20    63.42    63.01    58.33    62.16    61.22    57.95    59.64
HEAT INPUT TO FREE BLOCKS FROM GROUND COIL AND OR TANK
      0.       0.       0.       0.       0.       0.       0.       0.       0.  621903.
 621022.  499832.  508945.  417600.  437925.  361766.  392025.       0.       0.       0.
      0.  229406.  336202.  211991.  301871.  212854.  293288.  210645.  276332.       0.
      0.       0.       0.       0.       0.       0.       0.       0.       0.       0.

TOTAL HEAT INPUT=  5933697.


                   TIME = 1000.0000
    TOUT        QSUM        QCALC       MDOT
 6.918E+01   5.749E+03   5.826E+03   2.035E+03


*** THIS IS GROCS SPEAKING
HOUR=     470.00               MONTH =       6             YEAR =       0
   69.63    68.63    67.54    62.99    68.25    68.29    68.18    67.93    62.82    69.90
   70.13    69.89    70.12    69.74    69.93    69.41    69.52    69.35    69.37    69.07
   68.50    69.44    68.74    69.43    68.73    69.13    68.36    68.70    67.99    59.29
   59.55    65.99    66.02    65.44    65.12    58.85    63.82    63.07    58.47    60.22
HEAT INPUT TO FREE BLOCKS FROM GROUND COIL AND OR TANK
      0.       0.       0.       0.       0.       0.       0.       0.       0.  684447.
 677283.  560248.  563575.  479568.  495651.  429610.  457734.       0.       0.       0.
      0.  288925.  415469.  269062.  378376.  275415.  376595.  280656.  365279.       0.
      0.       0.       0.       0.       0.       0.       0.       0.       0.       0.

TOTAL HEAT INPUT=  6997894.


                   TIME = 1200.0000
    TOUT        QSUM        QCALC       MDOT
 7.087E+01   4.517E+03   4.884E+03   4.000E+03


*** THIS IS GROCS SPEAKING
HOUR=     670.00               MONTH =       6             YEAR =       0
   71.10    70.08    68.94    64.34    69.31    69.39    69.34    69.14    64.20    70.47
   70.62    70.52    70.69    70.44    70.59    70.18    70.28    70.03    70.17    70.01
   69.56    70.04    69.43    70.12    69.50    69.92    69.25    69.58    68.95    60.25
   60.50    67.13    67.24    66.82    66.55    59.36    65.16    64.63 [illegible] [illegible]
HEAT INPUT TO FREE BLOCKS FROM GROUND COIL AND OR TANK
      0.       0.       0.       0.       0.       0.       0.       0.       0.  730443.
 718102.  602819.  600802.  522260.  533845.  477146.  502357.       0.       0.       0.
      0.  334638.  478400.  310277.  436932.  319704.  439729.  331209.  433456.       0.
      0.       0.       0.       0.       0.       0.       0.       0.       0.       0.

TOTAL HEAT INPUT=  7772120.
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                   TIME = 1400.0000
    TOUT          QSUM          QCALC         MDOT
  7.117E+01     3.330E+03     3.745E+03     4.000E+03


*** THIS IS GROCS SPEAKING
HOUR=      140.00             MONTH =          7            YEAR =          0
     72.43      71.36      70.16      65.62      70.19      70.28      70.25      70.00      65.49      70.91
     70.97      70.98      71.06      70.94      71.00      70.72      70.74      70.46      70.63      70.54
     70.14      70.42      69.89      70.52      69.98      70.39      69.00      70.09      69.54      61.19
     61.43      67.91      68.04      67.75      67.50      59.87      66.13      65.75      59.51      61.25
HEAT INPUT TO FREE BLOCKS FROM GROUND COIL AND OR TANK
        0.         0.         0.         0.         0.         0.         0.         0.         0.    763696.
   748470.    632481.    627169.    551559.    560344.    510678.    534405.         0.         0.         0.
        0.    371406.    530203.    341961.    483817.    353020.    489481.    370050.    487804.         0.
        0.         0.         0.         0.         0.         0.         0.         0.         0.         0.

TOTAL HEAT INPUT=  8356623.


                   TIME = 1600.0000
    TOUT          QSUM          QCALC         MDOT
  7.136E+01     2.562E+03     2.601E+03     4.000E+03


*** THIS IS GROCS SPEAKING
HOUR=      340.00             MONTH =          7            YEAR =          0
     73.67      72.54      71.27      66.81      70.96      71.05      71.04      70.89      66.69      71.28
     71.25      71.35      71.34      71.33      71.31      71.15      71.00      70.77      70.95      70.80
     70.54      70.70      70.21      70.00      70.31      70.71      70.18      70.44      69.94      62.11
     62.32      68.46      68.60      68.39      68.16      60.36      66.81      66.54      60.03      61.65
HEAT INPUT TO FREE BLOCKS FROM GROUND COIL AND OR TANK
        0.         0.         0.         0.         0.         0.         0.         0.         0.    786640.
   771027.    652140.    645981.    570814.    579067.    533750.    558087.         0.         0.         0.
        0.    402447.    574477.    367806.    523225.    379731.    530685.    401779.    533286.         0.
        0.         0.         0.         0.         0.         0.         0.         0.         0.         0.

TOTAL HEAT INPUT=  8810951.


                   TIME = 1800.0000
    TOUT          QSUM          QCALC         MDOT
  7.150E+01     2.002E+03     2.027E+03     4.000E+03


*** THIS IS GROCS SPEAKING
HOUR=      540.00             MONTH =          7            YEAR =          0
     74.51      73.43      72.18      67.91      71.64      71.72      71.71      71.59      67.81      71.60
     71.48      71.68      71.57      71.67      71.55      71.52      71.37      71.03      71.19      71.15
     70.85      70.91      70.46      71.02      70.56      70.94      70.46      70.71      70.25      62.99
     63.19      68.87      69.00      68.85      68.64      60.85      67.32      67.12      60.54      62.00
HEAT INPUT TO FREE BLOCKS FROM GROUND COIL AND OR TANK
        0.         0.         0.         0.         0.         0.         0.         0.         0.    800671.
   787131.    663348.    658798.    581693.    591793.    548071.    575249.         0.         0.         0.
        0.    429112.    613365.    389643.    557508.    401990.    566030.    428571.    572632.         0.
        0.         0.         0.         0.         0.         0.         0.         0.         0.         0.

TOTAL HEAT INPUT=  9165664.


                   TIME = 2000.0000
    TOUT          QSUM          QCALC         MDOT
  7.161E+01     1.559E+03     2.027E+03     4.000E+03
```
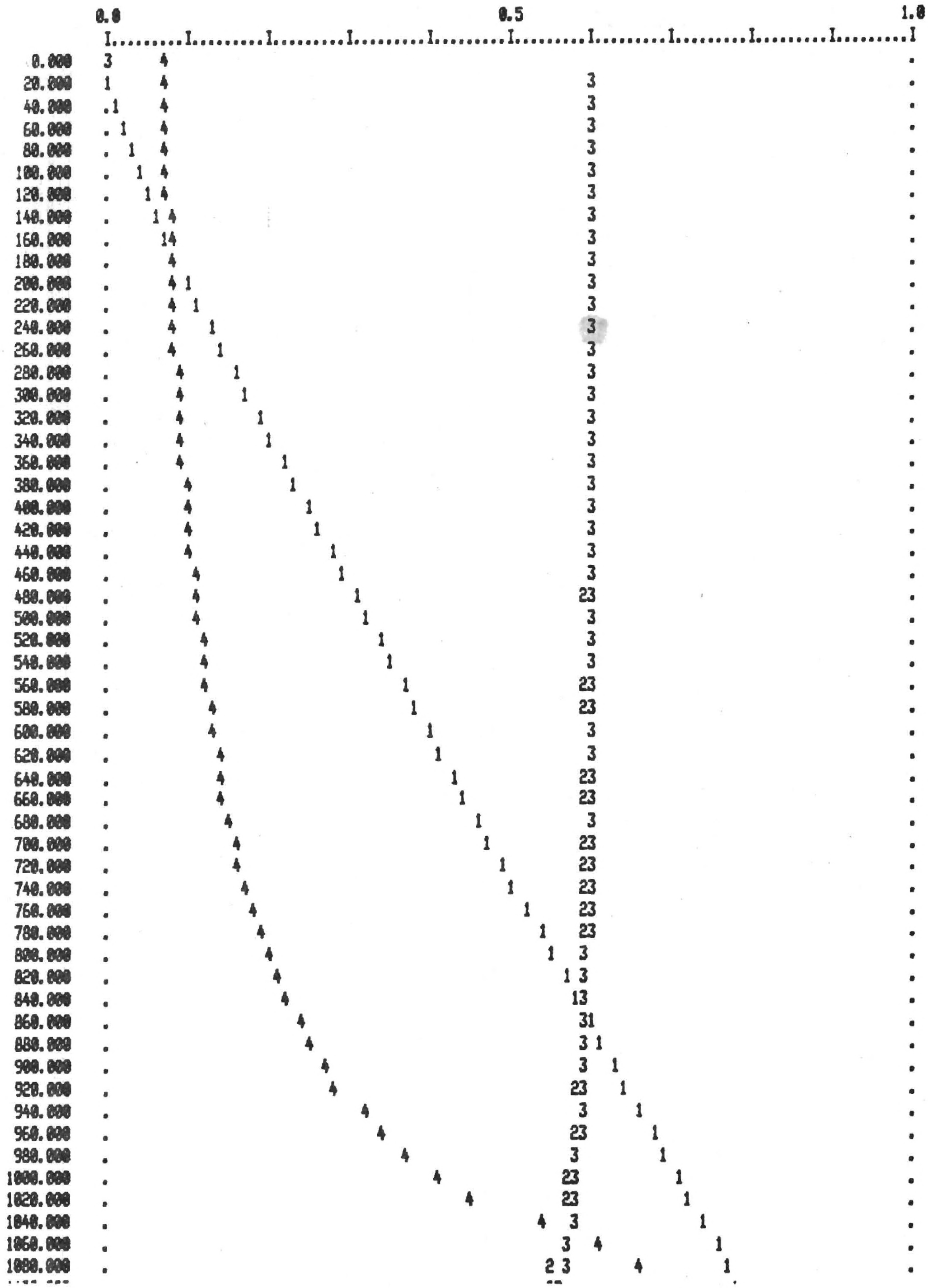
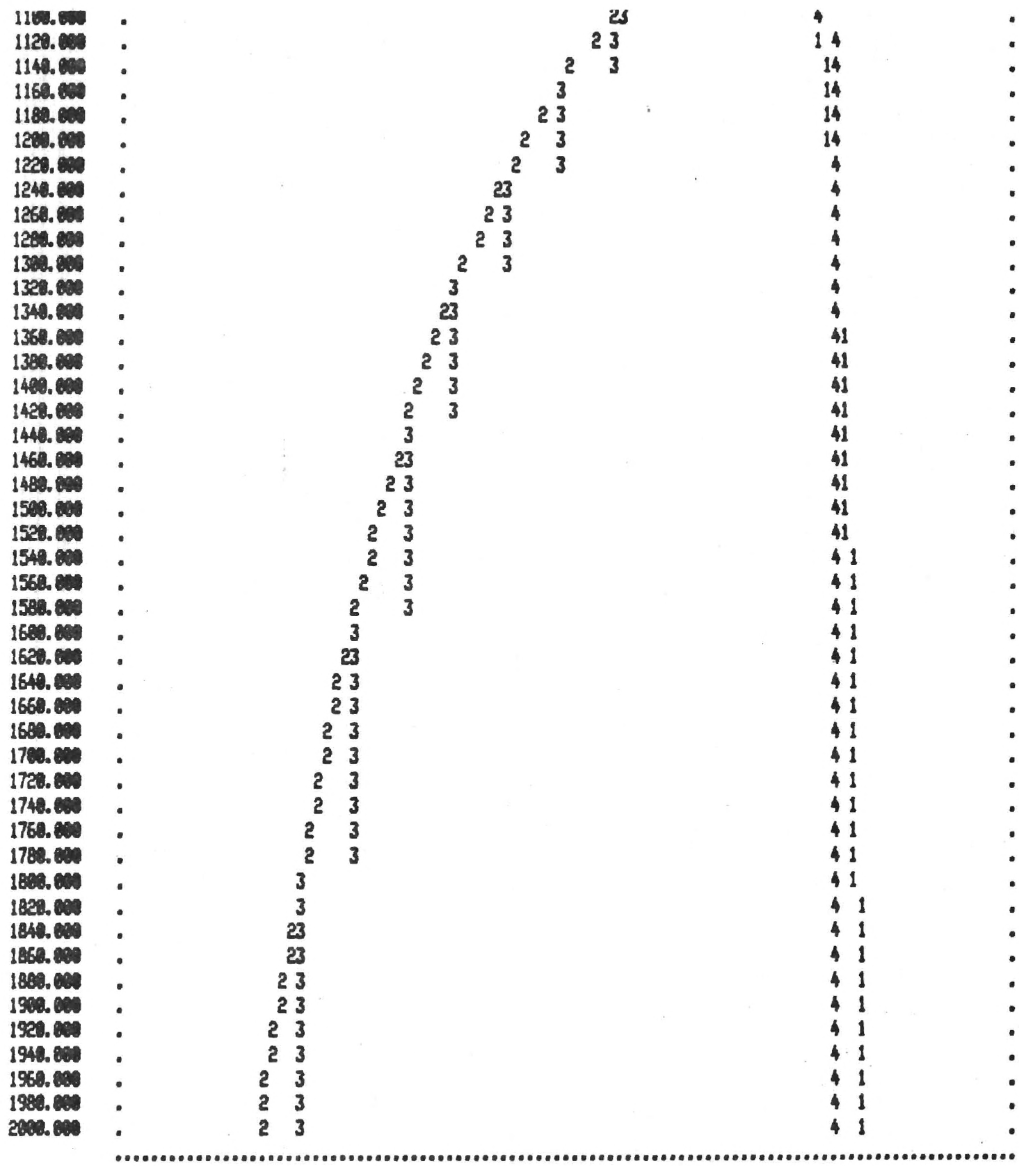

UNIT	1	WAS CALLED	126 TIMES
	2		1739
	3		133
	4		4121
	5		11
	6		101





SYMBOL	IDENTIFIER	SCALE FACTOR	ZERO POINT
1	TOUT	2.000E+01	5.500E+01
2	QSUM	1.000E+04	0.
3	QCALC	1.000E+04	0.
4	MDOT	5.000E+03	0.

87.149 CP SECONDS EXECUTION TIME.

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IDLE

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